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AIRCRAFT POWER PLANTS

Part I—Aircraft Engines

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Aircraft power plants.

[c1926]

AUTHORS' PREFACE

In preparing this volume, the authors have attempted to avoid duplicating the contents of numerous books on the market. The works now available on internal combustion engines are largely intended to meet the needs of those engaged primarily in the construction, development or maintenance of engines, and may be roughly divided into two classes—books suitable for engineers of considerable technical education, and those intended for readers of less technical training.

In the aeronautical field there is considerable need for information on the broad subject of engine characteristics and performance, by persons not primarily interested in engine design, development, or maintenance. This demand comes mainly from airplane and airship designers who must possess a general knowledge of the characteristics of the power plants with which they are working in order to adapt them successfully to the structural designs; also from pilots and operators who must be familiar with the general characteristics of the power plant so that they may intelligently and successfully operate the aircraft in which the power plant is installed. Ordinarily these readers have no intention of engaging actively in details of engine design; they are primarily interested in having a reasonable knowledge of the various engine types, and the general underlying facts which have led to development along existing lines.

In considering the requirements of this class of readers, it appears unreasonable to expect them to be familiar, or to be interested, in the mathematical treatment of thermodynamic problems, or in following through the mathematics of engine design and performance. On the other hand, a simple non-technical explanation of the fundamental thermodynamic principles applying to the selection and arrangement of types

with the factors affecting performance, not only is desirable, but will be of great value to persons concerned primarily in the application and operation of aircraft power plants. It is along these general lines that the present volume has been prepared, and it is for the class of readers described above that it is intended.

In arranging the text of Part I, Aircraft Engines, an effort has been made to compare the various types of heat engines from a thermodynamic point of view without introducing or requiring familiarity with the mathematical theory of thermodynamics. The advantages and disadvantages of the various types are discussed and the basic reasons underlying the development along existing lines have been brought out. In discussing engine design, mechanical details have been entirely disregarded and emphasis placed on general considerations of balance, cylinder arrangement, cooling, etc. In discussing engine performance and the factors affecting it, an effort has been made to give a satisfactory explanation of the various phases of the problem without resorting to mathematical discussion, and without going into excessive refinements which might tend to confuse rather than to clarify the problem.

The subject of Propellers, comprising Part II, is treated in a broad manner similar to Part I on Engines, except that the important working formulas used in propeller design are included to complete the explanation of design principles, but without the more complicated mathematics required for a complete development of propeller aerodynamics. However, these working formulas, with the accompanying graphs, are sufficient for designing the common forms of propellers within the limits of the coordinates given.

Part III, on the subject of Water Ballast Recovery, necessarily departs from the policy pursued in the preceding portions of the book, because this subject is very new, with its development just beginning. So little information is available that it is deemed advisable not only to set forth the thermodynamic

principles involved, but also in the final chapter, an example of a design problem for water ballast recovery equipment is solved in detail for the guidance of those who will continue the development of this essential auxiliary to the power plants of large airships.

June 10, 1926.

E. T. Jones
Robert Insley
F. W. Caldwell
Robt. F. Kohr

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AIRCRAFT POWER PLANTS

PART I
AIRCRAFT ENGINES

By

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INTRODUCTION

The aircraft power plant includes all devices used in the generation of power and its delivery to the airscrew. The problem of the airscrew and power required for various conditions of flight are considered separately in Chapters IX and X.

The engine is the heart of the power plant, for while it may not make up the major portion of the weight or bulk, it determines the character of all other items comprising the complete unit. The airship engine of today is an outgrowth of the airplane engine and owes its present state of development to the vastly greater expenditure of effort which the latter has received. This is not considered a misfortune, as much useless duplication of effort has been avoided by concentrating on the one type; practically all of the lessons learned in the development of this type are applicable to the other. However, it is evident that the two types should not be identical. It is highly probable that as the fundamental types of construction become stabilized, a more pronounced divergence in detailed arrangement will arise. The main lines on which this diversion is likely to proceed are suggested by certain differences in operating conditions. Chief among these are: first, the engine proper constitutes a smaller proportion of the total weight of an airship than of an airplane; second, the airship generally carries fuel for much longer duration than an airplane; third, while engine failure is a serious matter on an airplane, complete failure of the power plant is a much more serious matter on an airship; fourth, airship engines are generally accessible during flight, whereas airplane engines are rarely so. Thus, while light weight, low fuel consumption, reliability and accessibility are highly important for both the airship and the airplane, their relative importance varies considerably in the two cases. No definite order of importance for these desirable characteristics

can be given in either case since each individual problem must be considered by itself, and the importance attached to each characteristic will depend upon the nature of the service required.

It seems fairly evident that in general the airship engine will sacrifice weight for reliability and fuel economy to a greater extent than the airplane engine, while accessibility, particularly for minor repairs which might be made while aloft, will be given preference for airship service. Considering the power plant as a whole, the airship lends itself better to the development of the multiple engine system than does the airplane; this arrangement works out very neatly on a large airship. As the airship can maneuver at very low speed, it is not necessary to have the entire power plant under the immediate control of the pilot.

CHAPTER I

HEAT ENGINES

General Principles

The function of the heat engine is the transformation of heat into mechanical work. The law of conservation of energy, a fundamental principle of physics, states that energy cannot be created or destroyed. Hence in transforming heat into mechanical work, or work into heat, the total quantity of energy does not change. It is only the form in which energy appears that changes.

Units of energy. Since heat and work are both forms of energy, they might be measured in terms of the same unit. However, engineering practice has firmly established the custom of using different units for measuring heat and mechanical work. The unit of work is the Foot-Pound; that is, work is measured by the product of the force and the distance through which the force acts. The unit of heat is the British thermal unit (B.t.u.) which is defined as the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit. The relation between these units has been carefully determined and to the nearest whole number one B.t.u. is the equivalent of 778 foot-pounds. Since both units are measures of the same thing, that is, energy, the statement that one B.t.u. equals 778 foot-pounds is exactly equivalent to the statement that one meter equals 3.28 feet. In the former case, two units of energy are compared; in the latter case, two units of length.

Interconversion of heat and work. Heat and work being two forms of energy are interconvertible. However, certain conditions are necessary for conversion. On calling to mind the various examples of such conversion with which we are

familiar, it is at once evident that while work is very readily converted into heat under a wide variety of circumstances, it is only under certain very definite and generally artificial conditions that heat may be transformed into work. This difference is so pronounced that it might almost be said that the conversion of work into heat is a natural process while the reversal of this change is an artificial one.

Conversion of work into heat. It is scarcely necessary to point out illustrations of the conversion of mechanical work into heat. This conversion goes on continuously in all natural processes. A brief consideration will show that practically all of the mechanical work generated for power purposes is finally transformed into heat either directly, as in the case of friction, or indirectly, as in the case of electrical energy used for light and heat. It should also be noted that all losses through friction in any mechanical process illustrate the conversion of mechanical work into heat.

Conversion of heat into work. When we consider the conversion in the opposite sense, that is, the conversion of heat into mechanical work, it is at once evident that illustrations of this type of conversion are difficult to call to mind. For purposes of power generation, practically the only means of transforming heat into mechanical work is through the expansion of liquids, gases, or vapors. Other examples might be cited such as expansion of solids and the production of electrical energy by the thermo-couple which energy could be converted into mechanical work by various means.

Heat engine. The basic principle of operation of all heat engines is the same. A confined working substance either liquid, vapor, or gas, is subjected to a recurrent cycle of processes involving four more or less distinct phases: First, an increase in pressure brought about by mechanical means; second, the addition of heat accompanied by an increase in volume or pressure or both; third, an expansion; fourth, a re-

jection of heat bringing the working substance back to its initial condition. The fourth step may be accomplished either by abstracting heat from the working substance, or by rejecting the working substance and substituting a like quantity of working substance at a lower temperature. In either case the process is essentially one of heat rejection. The useful work available for power purposes has its origin in the fact that the working substance being expanded at a high temperature and compressed at a low temperature, the work derived from the expansion exceeds the work required for compression. This excess is available for useful power.

Exhaust losses. It should be carefully noted that, in order that this process may be repeated continuously it is essential that on the completion of expansion the working substance be cooled to the proper temperature for starting compression; otherwise the operating temperature would continuously build up from cycle to cycle, quickly reaching impractical values. As noted above, from the point of view of energy supply, it makes no difference whether this cooling is effected by actually abstracting heat from the working substance as is done in the condensing steam engine or by discarding the working substance and substituting a fresh supply of cold gas as is done in the internal combustion engine. In either case a considerable quantity of heat is rejected without being converted into useful work. This constitutes the major loss encountered in the conversion of heat into mechanical work. A study of the various possible working cycles conclusively demonstrates that it is impossible to make further use of the rejected heat in succeeding cycles, for the reason that the heat rejection takes place at a relatively low temperature, while the heat addition takes place at a high temperature. A basic law of physics states that heat will not flow from a low temperature source into a body at higher temperature, so that, even in the ideal case, a considerable portion of the heat supply is irretrievably lost. In

actual engines the quantity of heat rejected is usually considerably in excess of the heat that would be rejected by an ideal engine. For instance, in the gas engine it is theoretically possible to return to the cycle part of the heat lost, but practical reasons make it inadvisable to attempt this. Expansion could be carried further, thus reducing the heat loss, but this would necessitate an increase in the bulk and cost of the engine, and it would increase the friction loss so that in practice better results are obtained by not carrying the expansion too far. The efficiency of the gas engine could be increased by arranging to transfer heat from the exhaust gas to the fresh charges after completion of the compression stroke. However, the mechanical equipment required to affect this transfer would completely offset the slight gain in economy effected. The important points to be noted in connection with the exhaust loss are: (1) even in the ideal case this loss constitutes a considerable portion of the heat applied, and (2) that in practical engines this loss is further increased in order to reduce bulk, cost and complication.

Heat engine efficiency. The efficiency of the heat engine is the percentage of the total heat supply which is converted into useful work. Since the operation of the heat engine is dependent on the rejection of a considerable portion of the heat supply the efficiency cannot approach 100 per cent, even when all losses due to friction, leakage, radiation, etc., are eliminated. In this respect the heat engine differs from most machines in which the losses are incidental and may be reduced by careful design. The largest loss in heat engines is the heat rejected at the end of the expansion and this loss is an essential part of the process of power generation. To this loss must be added all the losses which, while not essential to the process, are unavoidable. These include friction, leakage, loss of heat by radiation and conduction. When all of these losses have been considered it will be found that the efficiency of a heat

engine seldom exceeds 30 per cent, whereas the efficiency of a machine for converting electrical energy into work may easily exceed 95 per cent.

Figure 1 illustrates the disposition of the heat which is characteristic of all heat engines. The exact percentage of the total heat represented by each part of the diagram varies considerably in different types of engines. The figures given are

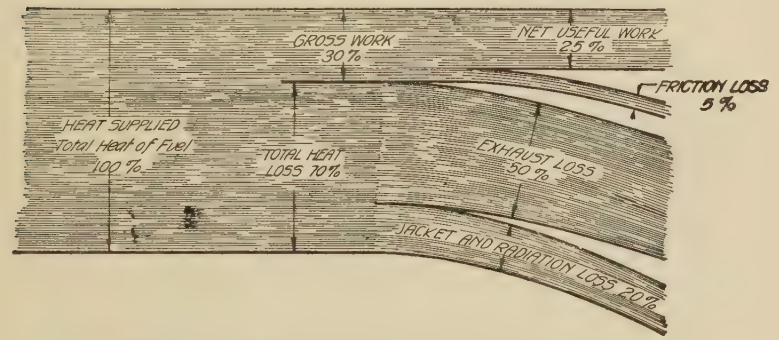


Figure 1. Heat Distribution in Typical Heat Engine

intended to represent a general average for fairly efficient heat engines.

Classification of Heat Engines

Practical power plants may be divided into two classes—the external combustion type and the internal combustion type. In the former class we have the steam engine, the steam turbine, and mercury vapor turbine; in the latter we have the gas engine, the oil engine, and the gas turbine.

Steam engine. The steam engine need hardly be considered for aircraft purposes, as it has been made almost obsolete, so far as the economical generation of power is concerned, by the steam turbine which is more efficient, lighter, more reliable, and more compact than the steam engine. Comparing the steam engine with the gas engine, it is evident that since the

steam engine requires a number of fairly heavy accessories, such as boilers, pumps, fireboxes, blowers, etc., the engine itself would have to be much lighter than a gas engine of the same power. The weight of the engine proper is largely a matter of average effective cylinder pressures and speeds of revolution. When operating under conditions giving best economy, the average effective pressure in the steam engine cylinder is much lower than that in the gas engine cylinder; hence the displacement of the steam engine must be correspondingly greater than that of the gas engine so that even operating on the two-stroke cycle and using the double acting principle, the steam engine can not be built at weights sufficiently below those of the gas engine to compensate for the weight of the accessories which the steam engine requires.

Steam turbine. The chief advantages of the steam turbine for aircraft power plants are: great reliability, ability to utilize heavy oil as fuel, the fact that its output is not affected by altitude, and the smoothness of its operation. The principal disadvantages are its great complication and probably its weight. A steam turbine power plant would necessarily include the following items: a blower with driving motor and means of regulation, a furnace with proper heat insulation and supports, a fuel pump with drive and regulation, a boiler, a superheater, the turbine, a condenser, feed pump with drive, and an air pump. For most economical operation, feed water heaters and economizers would also be required. Figure 2 indicates diagrammatically the essential elements of a steam turbine power plant with lines indicating the flow of the working substance and the furnace gases through the various elements. The control of such a power plant would be highly complicated compared with the conventional gas engine, and would require the undivided attention of at least one man. It is difficult to estimate the weight of a steam turbine power plant designed for aircraft use since nothing approaching such

a design has ever been built, but it is very doubtful whether a unit of this type could even approach the modern aircraft engine either in bulk or weight. From the point of view of fuel economy, the advantage lies with the internal combustion engine and this advantage is especially true when operation over a fairly wide range of power output is required. It should further be noted that while numerous schemes have

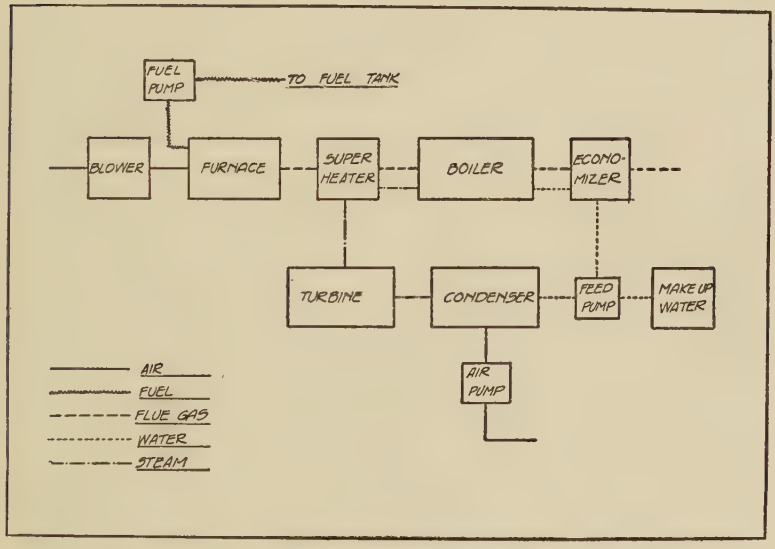


Figure 2. Elements of Steam Turbine Power Plant

been developed for improving the economy of the steam turbine, all of these plans require additional complication and additional weight so that while they may be very advantageous for stationary power plants, much, if not all, of the advantage of these devices would be lost in an aircraft power plant.

Mercury vapor turbine. The mercury vapor turbine power plant is in reality a combination of a mercury turbine power plant and steam turbine power plant. This type is undoubtedly heavier than the steam power plant for given output and is

also considerably more complicated. Its only advantage over the steam power plant lies in its superior fuel economy and it is doubtful whether the mercury vapor power plant will show economy superior to the best that may be obtained from internal combustion engines.

Internal combustion turbine. The internal combustion turbine need not at present be considered as a practical aircraft power plant for the reason that nothing approaching a light weight, economical power plant of this type has yet been produced and there appear to be certain fundamental difficulties in connection with its development, a solution of which is by no means in sight. Chief among these difficulties are the lack of a rotary compressor which can compare in efficiency with a reciprocating compressor, and the lack of materials which will successfully operate at the high temperatures and stresses required in a turbine of this type. Efforts to reduce the working temperature of the gases and thus reduce the working temperature of the turbine wheel consist essentially in wasting a considerable portion of the heat available. This necessarily results in a reduction in fuel economy which is fatal to an aircraft engine.

Internal combustion engine types. Internal combustion engines may be divided into two classes commonly referred to as gas engines and oil engines. The former includes all engines which utilize a fuel sufficiently volatile to be efficiently introduced into the cylinder by mixing with the entering air. The latter class includes all engines which utilize liquid fuels of low volatility which must be introduced into the engine by spraying directly into the combustion chamber of the cylinder. Another important difference between the two types is to be found in the method of ignition: gas engines usually require some form of electric spark ignition, while the oil engines usually rely upon the heat developed during compression for igniting the fuel. A further difference between the two types

is to be found in the time at which combustion is started and the rate at which the combustion proceeds after starting. In the gas engine the charge is usually ignited well before top dead center and an effort is made to secure very rapid combustion of the entire charge. In the oil engine ignition usually takes place at or slightly before top dead center and the rate of combustion is controlled by the rate at which fuel is injected into the cylinder from the injection nozzle. In the earlier types this rate was usually so adjusted that the pressure remained approximately constant during the period of combustion, while in the more modern high speed types employing solid injection the rate of combustion approaches that of the gas engine.

Two-cycle and four-cycle types. Both the gas engine and the oil engine have been developed in two forms known as the four-stroke type and the two-stroke type. In the former, one entire stroke of the piston is devoted to exhausting the used charge after expansion and a second entire stroke is utilized in drawing in a fresh charge. In the two-stroke type the fresh charge is introduced into the cylinder at the end of the expansion stroke, driving the exhaust out before it, both inlet and exhaust valves being open at the same time. The two-stroke type has the advantage of giving a power impulse for every revolution, whereas the four-stroke type gives a power impulse for every two revolutions. For this reason the two-stroke type is very attractive where maximum power output, combined with minimum weight, is desired.

On the other hand, the very short time available for exhausting the old charge and introducing the fresh one results in these functions being carried out in a very incomplete manner in high speed engines. This results in a reduction of power output and efficiency. To date, gas engines of the high-speed four-stroke type have shown superior fuel economy and greater output per unit weight, combined with greater ease of control

than has been possible in the two-stroke type. As regards the oil engine, both the four-stroke and two-stroke type have been developed to about the same stage of performance and it seems quite probable that in this class of engine the two-stroke type may have the margin of advantage over the four-stroke type.

Oil engine. As indicated above, the oil engine is, in most respects, similar to the gas engine and possesses most of the advantages of the gas engine as an aircraft power plant. When compared with the gas engine it possesses the distinct advantage of using a fuel which is not highly volatile and thus does not constitute a serious fire risk. It has the further advantage of not requiring electrical ignition, which is always a more or less troublesome item on the gas engine. The major disadvantage of the oil engine lies in its greater weight and the inherent difficulty in starting. The fundamental reason for the excessive weight of the oil engine is to be found in the method of combustion. The cylinder is charged with air which is compressed to a very high pressure, usually in the vicinity of 600 pounds per square inch. This results in a rise in temperature sufficient to cause ignition of the fuel when it is sprayed into the combustion chamber at the end of the compression stroke. Combustion starts immediately at the beginning of injection and continues throughout the injection period. There is thus no opportunity to mix the fuel and air thoroughly before combustion is started and this results in a rather slow rate of combustion, and requires a considerable excess of air in order that combustion may be complete. This limits the operating speed of the engine and increases the cylinder size due to the excess air required. Attempts to increase the speed of the engine or reduce the quantity of excess air result in incomplete combustion which reduces the cycle efficiency and tends to foul the cylinder with partially burned fuel. The speed of the oil engine being thus limited, this type lends itself better to development in the two-stroke type than does the

gas engine, the speed of which is not restricted by rate of combustion. It is probable that further development will result in a fairly satisfactory oil engine for aircraft use and it is predicted that the engine will be of the two-stroke type. The chief problems in connection with this development appear to lie in the development of an efficient system of induction and exhaust for the two-stroke cycle and acceleration of the combustion process by better mixing of the fuel and air, thus speeding up combustion and reducing the quantity of excess air required.

Gas engine. The gas engine possesses, to a remarkable degree, those qualities which are desirable in an aircraft engine. The chief advantages of this type for aircraft service are low weight per horsepower, high thermal efficiency, compact arrangement, simplicity of control, ease and quickness of starting, maintained efficiency over a wide range of power output, and freedom from complicated or bulky accessories. Its chief disadvantages are the necessity of using a highly volatile fuel which constitutes a serious fire risk, rapid loss of power at higher altitudes, and rather complicated mechanical construction. The low unit weight of this type of engine results from its ability to operate efficiently at high crankshaft speeds and the high average effective cylinder pressure developed, this low weight factor being better than that obtained in any other reciprocating engine. The excellent economy of the gas engine, compared with external combustion engines, may be attributed to the entire absence of boiler and furnace losses, as the combustion takes place within the engine cylinder.

Comparing the gas engine with the oil engine, its thermal efficiency is slightly lower than that of the low-speed oil engines, but is considerably higher than that of high-speed oil engines which have been developed to date. The chief source of mechanical complication of the gas engine lies in the necessity of utilizing multiple cylinders. This is necessary in order

to smooth out the uneven torque developed in a single cylinder and to secure satisfactory inertia balance without the use of heavy balance weights. The accessories employed with a gas engine are much more simple than those used on any type of external combustion engine, and are probably not more complicated than those required for the oil engine; the necessity for electrical ignition on the gas engine being offset by the complication of numerous injection pumps and nozzles on the oil engine. To date, the advantages listed above have been sufficient to make the gas engine supreme in the aeronautical field. No other type has as yet been developed which even remotely approaches the gas engine when considered from all points of view. So far as the writer is aware, no aircraft has ever been successfully flown using any other type of engine. For this reason the remainder of this discussion will deal exclusively with the aircraft power plant as developed around the gas engine, no further reference being made to either the external-combustion type or the oil engine.

Comparison of engine performances. The following table gives a comparison of the thermal efficiencies of the various types of power plants discussed above. The values given are representative of what may be expected from high-grade units in everyday service, rather than the most favorable results that have ever been obtained.

	Per Cent
Steam engine (uniflow or triple expansion)	22
Steam turbine	25
Diesel engine (low speed)	32
Gas engine (aviation 5.5 compression)	25
Gas engine (aviation 7.5 compression)	31
Mercury turbine	32

CHAPTER II

ENGINE PERFORMANCE

Definitions

Any discussion of internal combustion engine performance must employ a number of terms which require definition.

The *brake horsepower* of an engine is the horsepower available for propulsion—the useful power delivered to the propeller.

Corrected brake horsepower is a theoretical horsepower computed for comparative purposes, representing the horsepower that the engine would develop under standard atmospheric conditions (usually a barometric pressure of 29.92 inches of mercury and a temperature of 60° F.). It is customary to calculate the corrected brake horsepower on the assumption that the horsepower varies directly as the barometric pressure and inversely as the square root of the absolute temperature of the air entering the carburetor.

The *indicated horsepower* is the horsepower actually developed in the cylinders. It exceeds the brake horsepower by the amount of the *friction horsepower* which is the power required to overcome the internal friction of the engine including the power required to pump the fuel mixture into, and the exhaust gases from, the cylinder (the pumping loss).

The *mechanical efficiency* is the ratio (expressed ordinarily in per cent) of the power developed to the power available, or the ratio of brake to indicated horsepower.

Volumetric efficiency (expressed in per cent ordinarily) is the ratio of weight of air admitted to the cylinders to the weight of a volume of air equal to the piston displacement under atmospheric conditions.

The *torque* of the engine is the turning moment of the engine (usually expressed in pound-feet).

Indicated mean effective pressure is the pressure developed in the cylinder during one cycle averaged over one stroke. It is purely a figure of merit computed from the power developed but representing no actual pressure in the cylinder.

The *brake mean effective pressure* is that part of the indicated mean effective pressure representing the brake horsepower of the engine.

The *friction mean effective pressure* is that part of the indicated mean effective pressure representing the friction horsepower of the engine including the pumping loss.

The *rate of fuel consumption* is ordinarily expressed as a ratio of fuel consumption to horsepower in pounds of fuel per horsepower per hour. When so expressed it is called *specific fuel* consumption and may be computed as either brake or indicated specific fuel consumption using either brake or indicated horsepower in the ratio.

Oil consumption is similarly expressed, as pounds per horsepower per hour.

The *thermal efficiency* of an engine is the ratio of power delivered to potential power input (in the heat value of the fuel).

The *compression ratio* of the engine is the ratio of maximum to minimum cylinder volume or the total volume to the compression volume.

The *air standard efficiency* is a purely theoretical term representing the ratio of the maximum power obtainable from the fuel (with the compression ratio and working cycle employed) to the power available in the fuel used.

The *relative efficiency* is the ratio of the power obtained to the power theoretically obtainable, or the ratio of the thermal to the air standard efficiencies.

The above terms are also subject to mathematical definition:

Torque (T) in pound-feet

T = brake load (pounds) $\times$ brake arm (feet)

Brake horsepower (B.hp.)

$$\text{B.hp.} = \frac{2\pi TN}{33{,}000}$$ (where N is the engine speed in revolutions per minute)

Corrected brake horsepower (Hp.)

$$\text{Hp.} = \text{B.hp.} \times \frac{29.92}{B} \times \sqrt{\frac{t}{520}}$$ (where B is the observed barometric pressure and t the observed absolute temperature in degrees Fahrenheit)

Friction horsepower (F.hp.)

$$\text{F.hp.} = \frac{2\pi T_f N}{33{,}000}$$ (T_f is friction torque)

Indicated horsepower (I.hp.)

$$\text{I.hp.} = \text{B.hp.} + \text{F.hp.}$$

Mechanical efficiency (E_m) *per cent*

$$E_m = \frac{\text{B.hp.}}{\text{I.hp.}} \times 100$$

Brake mean effective pressure (*BMEP*) in pounds per square inch

$$BMEP = \frac{792{,}000 \times \text{B.hp.}}{D \times N}$$ (where D is the total piston displacement in cubic inches)

Indicated mean effective pressure (*IMEP*) pounds per square inch

$$IMEP = \frac{792{,}000 \times \text{I.hp.}}{D \times N} \text{ or } \frac{BMEP \times 100}{E_m}$$

Friction mean effective pressure (*FMEP*) pounds per square inch

$$FMEP = IMEP - BMEP \text{ or } \frac{792{,}000 \times \text{F.hp.}}{D \times N}$$

Brake specific fuel consumption (C_s) pounds per horsepower per hour

$$C_s = \frac{C}{\text{B.hp.}}$$ (where C is hourly fuel consumption in pounds)

Indicated specific fuel consumption (C_{si}) pounds per horsepower per hour

$$C_{si} = \frac{C}{\text{I.hp.}}$$

Specific oil consumption (OC_s) pounds per horsepower per hour

$$OC_s = \frac{OC}{\text{hp.}}$$ (where OC is the hourly oil consumption in pounds)

Thermal efficiency E_t (brake) or E_{ti} (indicated) in per cent

$$E_t = \frac{2{,}545}{C_s \times H} \times 100$$ (where H is the heat content of the fuel in B.t.u. per pound)

$$E_{ti} = \frac{2{,}545}{C_{si} \times H} \times 100$$

Air standard efficiency (E) in per cent

$$E = \left[1 - \left(\frac{1}{R}\right)^{.408}\right] \times 100$$ (where R is the compression ratio)

Relative efficiency—per cent

$$\text{Relative efficiency} = \frac{E_t}{E} \times 100$$

The *weight-horsepower ratio,* expressed in pounds (dry weight) per normal brake horsepower, is a measure in a general sense of the suitability of the engine for aircraft use.

Factors Affecting Performance

Full throttle power output of a gas engine depends upon cylinder performance and crankshaft speed. The overall cylinder performance is measured by the indicated mean effective pressure. Figure 3 shows the full throttle performance of a typical aircraft engine. Curves are given of brake horsepower, brake fuel consumption, friction loss, indicated mean effective pressure and brake mean effective pressure.

Speed limitations. Reference to this diagram shows that the power output of the engine continues to increase as the speed is increased up to values considerably above the normal speed. This is due to the fact that mechanical considerations impose a limiting speed lower than the speed at which maximum horsepower is developed. This is typical of all modern engines. The stresses set up in the moving parts and the severity of the bearing loads are largely determined by the inertia forces set up by the reciprocating parts. These forces vary as the square of the crankshaft speed; hence, rise rapidly as the speed is increased. For this reason the maximum safe speed of operation is largely dependent upon the weight of the reciprocating parts, and is

usually well below the speed at which maximum power may be developed.

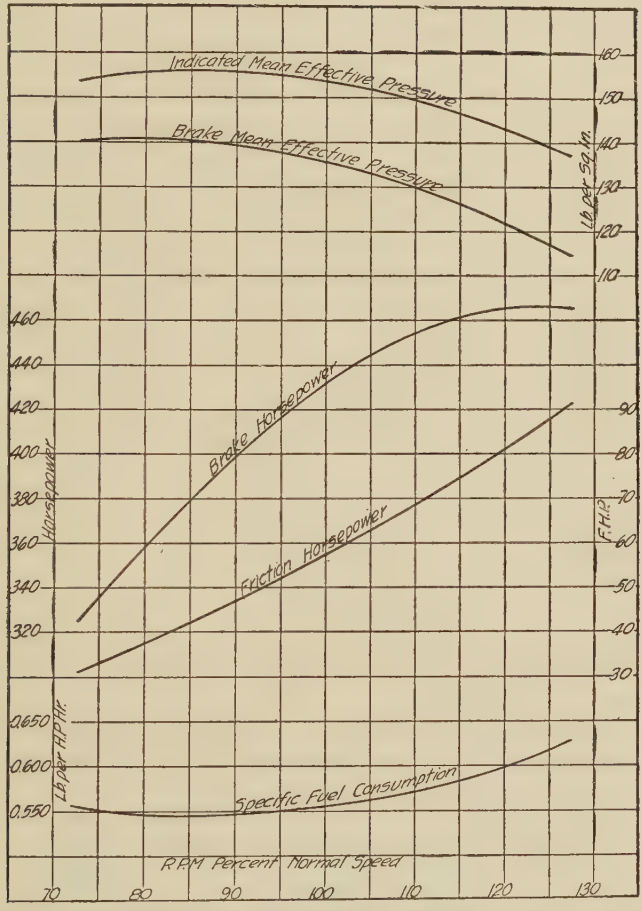


Figure 3. Full-Throttle Performance of Typical Aircraft Engine

Friction horsepower. Friction horsepower includes both pumping losses and mechanical friction caused by bearings, accessories, valve gear, etc. The mechanical friction loss varies approximately as the crankshaft speed, while the pumping loss

increases more rapidly; hence, the friction loss curve has a slight upward curvature.

Indicated mean effective pressure. The curve of indicated mean effective pressure (see Figure 3) rises at the lower speeds, reaches a maximum and begins to fall off again at the higher speeds. The point at which the curve is maximum is mainly affected by the volumetric efficiency of the engine, which is in turn dependent upon the valve timing, valve area and the design of the induction system. The relation between volumetric efficiency and r.p.m. is far from simple. In general, the pressure losses in the induction system increase as the speed increases. This tends to reduce the volumetric efficiency. On the other hand, the heat absorbed by the air in passing through the induction system is decreased at high speed which tends to increase the weight of air entering the cylinder. Pulsations in the manifold due to the uneven rate at which the air is drawn in, have a very pronounced effect on the volumetric efficiency and the selection of a valve timing which makes use of these pulsations in filling the cylinders, usually results in maintaining high efficiency up to comparatively high speeds.

Brake mean effective pressure. Brake mean effective pressure is the indicated mean effective pressure corrected for the friction loss. The curve, in general, follows that of the indicated mean effective pressure except that it reaches a maximum sooner and falls off more rapidly at high speeds due to the upward curvature of the friction horsepower curve. It is desirable that the maximum point of the brake mean effective pressure curve occur at approximately normal speed. However, this condition is usually difficult to obtain in a high-speed engine, because it is difficult to so design the induction system as to secure maximum volumetric efficiency at this point. It will be noted that this curve falls off rapidly at speeds considerably above the speed at which the maximum value occurs and that a point is finally reached where the drop in brake mean effective

pressure exactly offsets the gain in power due to increase in speed. This point corresponds with the maximum of the brake horsepower curve. This constitutes a second limitation of the maximum speed of the engine but, as stated above, in the present state of the art the mechanical stresses usually impose a speed limit before this point is reached.

Specific fuel consumption. The variation of specific fuel consumption with speed is largely affected by the characteristics of the carburetor. The curve shown in Figure 3 is based on the assumption that the carburetor is readjusted for best operation at each speed; therefore it represents the performance of an ideal carburetor. The curve of specific fuel consumption has a minimum in the vicinity of the maximum point of the brake mean effective pressure curve. The tendency of the curve to rise as the speed is increased beyond this point may be attributed to the increase of the friction loss of the engine and the reduction in *BMEP,* while the higher fuel consumption at lower speed is due largely to higher thermal losses resulting from the hot gases being in contact with the cylinder walls for a longer period at low speed. Leakage losses also become more important at low speed. The reader should bear in mind that these curves are subject to wide variations in individual engines due to peculiarities of the carburetor and variations in the distribution system of the engine. No safe rule for estimating the behavior of individual engines can be established; for reliable data, tests of the particular engine in question are necessary.

Propeller load. As actually operated in aircraft, the entire power of the engine is absorbed by the propeller and the characteristics of the propeller determine the actual power required of the engine at any given speed. The relation between the power required and speed for an airscrew closely follows the cube law, being somewhat dependent upon the rate of advance. The propeller is usually so designed that the power required

equals the full throttle power of the engine at normal speed. This is shown in Figure 4. The upper power curve indicates the full throttle output of the engine and the lower power curve

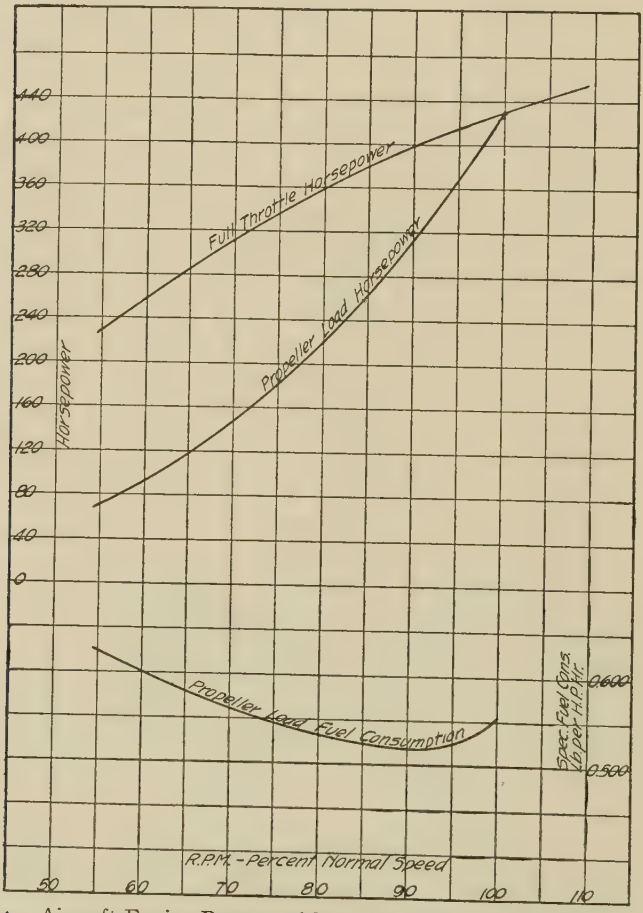


Figure 4. Aircraft Engine Power and Fuel Consumption with Propeller Load

is a cube curve meeting the full throttle curve at normal speed. In actual operation this curve is not definitely fixed but varies somewhat with the speed of advance. A higher speed of advance would move the curve to the left, while a lower speed

of advance would move it to the right. In general, this variation does not exceed 10 per cent throughout the normal working range.

Fuel consumption on propeller load. Referring to Figure 4, it will be noted that the curve of fuel consumption on propeller load differs considerably from that shown for full throttle operation. This difference is due mainly to the variation in mechanical efficiency with throttled operation and the metering characteristics of the carburetor. Since the friction loss depends mainly upon the r.p.m., the mechanical efficiency is lower at part throttle operation than at full throttle. This tends to increase the specific fuel consumption as the speed is reduced on propeller load. Individual carburetor characteristics vary widely, but, in general, the mixture tends to lean out at first as the throttle is closed and then to rapidly become richer as the fully closed position is approached. These characteristics are varied by the metering system used in the carburetor and the arrangement of the induction system. However, the curve shown in the diagram is considered typical of a modern engine. When exact data are required there is no safe rule to be followed; actual tests of the particular engine and carburetor combination are necessary.

Fuel air ratio. While the gas engine is capable of operating on mixtures of fuel and air, the proportions of which vary through a rather wide range, the range through which maximum power accompanied by good economy can be secured is rather narrow. Figure 5 indicates the effect of varying mixture ratio on *MEP* and specific fuel consumption. This diagram is considered typical of aviation engines using domestic aviation gasoline as fuel.

The limiting fuel to air ratios which will give satisfactory running are fixed by the fact that beyond these values the charge does not ignite and burn regularly. Starting with the leanest mixture which will give smooth and regular operation,

the power rises rapidly at first and then more gradually until a maximum is reached. Past this point the power falls off gradually until the other extreme of satisfactory operation is reached. It will be noted that the maximum output corresponds to a

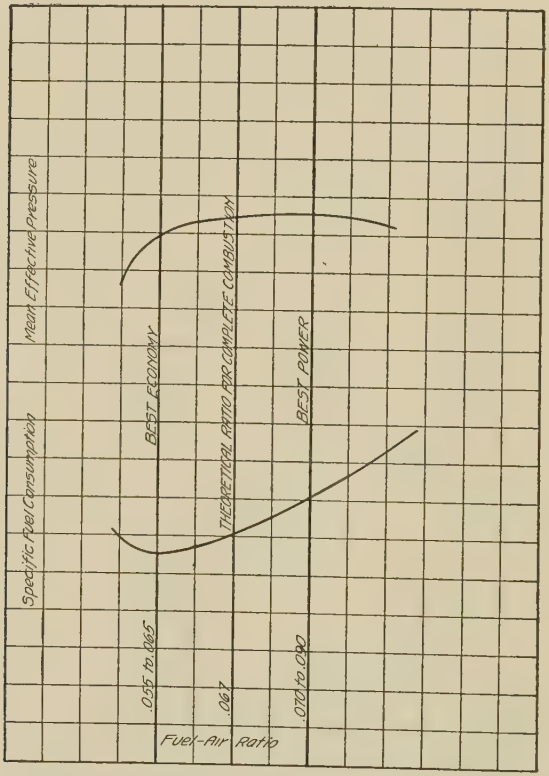


Figure 5. Effect of Fuel-Air Ratio Variation on Engine Performance

richer mixture than that which gives best economy. It should also be noted that the theoretically correct mixture, that is, mixture having exactly the quantity of air necessary to burn the fuel present, lies about midway between the mixture giving the maximum output and that giving best economy. As stated above, the actual proportions of fuel and air which enter the

engine cylinder are controlled by the metering characteristics of the carburetor, and while approximately constant mixture ratios are desired, it is usually impossible to hold this ratio absolutely constant so that the mixtures received by the engine vary considerably as the speed and load are varied. It is, however, usually possible in aircraft engines to maintain the proper mixture ratio within reasonable limits throughout the range from 50 to 100 per cent normal power.

Distribution. While the fuels used in gas engines are ordinarily referred to as volatile fuels, none of the fuels in ordinary use are sufficiently volatile so that complete vaporization takes place before the mixture enters the engine cylinder. This results in the mixture passing from the carburetor to the engine cylinder with a considerable portion of the fuel in a liquid state. The function of the intake manifolds is to distribute this mixture uniformly to the various cylinders, but this function is seldom carried out in a completely uniform manner. The liquid particles tend to resist sudden changes in direction of flow, such as are set up by bends, and tend to accumulate on the walls of the manifold and to flow along these walls in small streams, the result being that certain cylinders usually receive a larger portion of the fuel than others, resulting in a tendency toward uneven operation due to the variation in the fuel air ratio of the charge drawn into the various cylinders.

Referring to Figure 5, it will be noted that when operating in the vicinity of mixture ratio giving best economy, the power output is affected to a much greater extent by a scarcity of fuel than by an excess of fuel. For this reason, smooth operation requires that sufficient fuel be supplied to the cylinder receiving the leanest mixture to permit this cylinder to develop nearly maximum power. This, of course, results in a considerable excess of fuel going to those cylinders which receive the richest mixture and raises the average fuel consumption of the entire group. This is probably the most serious disadvantage

of uneven fuel distribution, although other effects are present such as variations in mean cylinder temperature and tendency towards carbon formation. Since the chief difficulty in securing uniform distribution is to be traced to the presence of liquid fuel in the manifolds, distribution may be improved by reducing the quantity of liquid fuel to a minimum. Application of heat to the intake manifolds helps materially in this respect, for which reason, intake manifolds are usually water-jacketed, although application of heat to the in-going charge has an adverse tendency on volumetric efficiency and tends to lower the power output of the engine. The same result may be secured by the application of heat to the air before entering the carburetor.

Effect of air temperature on power output. The power developed in the engine depends primarily on the weight of mixture used. The density of the entering air is a function of its temperature, hence at higher temperatures the weight of mixture entering the engine cylinder is reduced. This results in a reduction of power as the temperature of the entering air is increased. Considering the matter in this light alone, it would appear that the power should vary inversely as the absolute temperature of the entering air. However, the phenomena are complicated by the fact that considerable quantities of heat are added to and taken from the entering air as it passes from the air intake through the carburetor and manifold into the engine cylinder, the principal effects being a loss of heat at the carburetor due to vaporization of a portion of the fuel, loss of heat due to vaporization of the fuel in the intake manifold, heat abstracted from the water jackets surrounding the manifold walls and the heat absorbed in the engine cylinder during the induction stroke by contact with the residual exhaust gas and the hot cylinder walls. The exact amount of heat transferred, due to each of these causes, is to a certain extent dependent upon the temperature at which the air enters the sys-

tem. Therefore, it is evident that there is no simple definite relation between the temperature of the entering air and the power output of the engine which holds true of all engines under all conditions of operation.

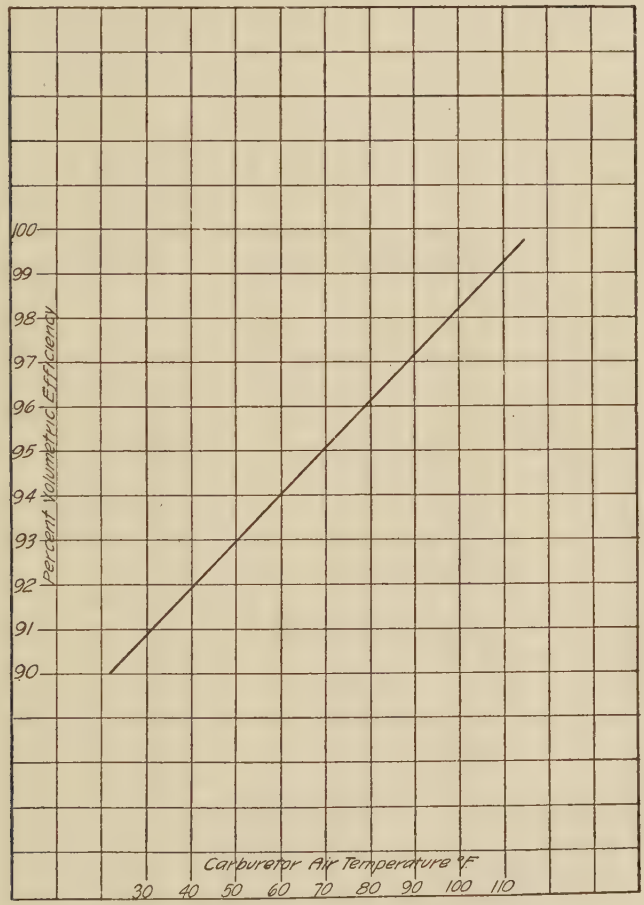


Figure 6. Effect of Air Temperature on Volumetric Efficiency

The effect of air temperature on volumetric efficiency has been determined experimentally for a number of American aviation engines and the curve shown in Figure 6 is typical. It

will be noted that the increase in volumetric efficiency resulting from an increase in air temperature tends to offset the reduction in air density. As a rule the variation in power output

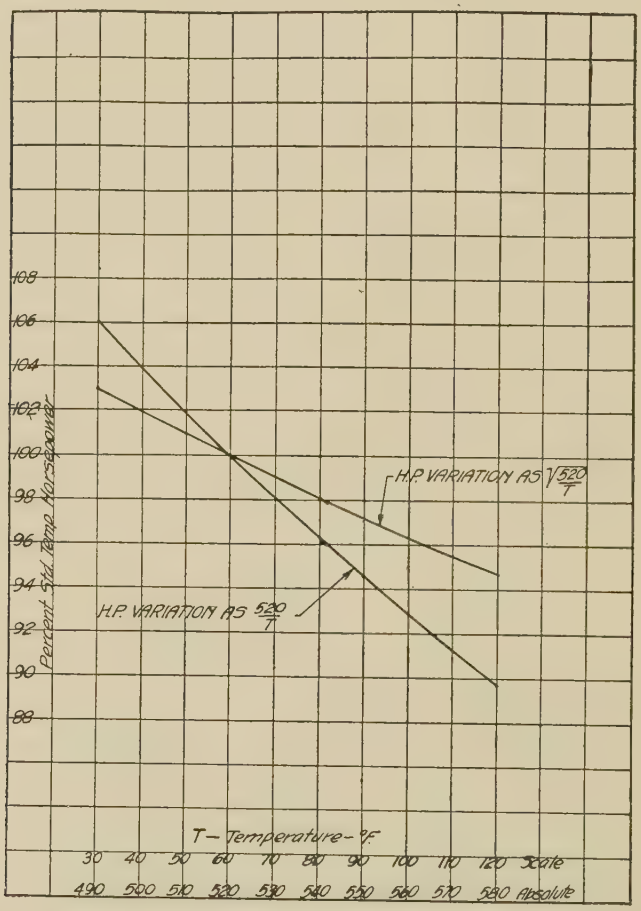


Figure 7. Curves of Air Absolute Temperature Ratios for Test Result Corrections

with air temperature is less rapid than would be expected from considerations of density alone. As a result of a number of careful tests conducted by the Bureau of Standards, the Army

and Navy Air Services have adopted an empirical law to be used in correcting engine horsepower for variations in temperature. Sixty degrees F. (60°) has been selected as the standard temperature of reference, and it is assumed that the power varies inversely as the square root of the ratio of the absolute temperature at time of test to the standard absolute temperature. This law is recognized as being only approximate and should be applied only over a rather narrow range of temperature, as from 32° to 120°F. Figure 7 shows this relation; the curve indicating a power variation proportional to density is drawn in for comparison.

Jacket water temperature. Variations in the temperature of the jacket water affect the performance of the engine in a number of ways. Increasing the jacket temperature increases the heat absorbed by the incoming charge which lowers the volumetric efficiency. The viscosity of the lubricating oil is reduced, which reduces the friction loss. These effects are illustrated in Figures 8 and 9, both of which are taken from actual tests. The extent to which these effects are present in any particular engine depends on a great many features of the design and cannot readily be estimated. In general, it may be said that in engines which are well cooled an increase in jacket water temperature gives a slight increase in power output, while in engines whose cooling is doubtful, particularly as regards pistons, valves and spark plugs, an increase in jacket temperature results in a loss of power.

Spark advance. While combustion in a gas engine is very rapid, it is by no means instantaneous. In a high-speed engine the time required for combustion represents a considerable portion of the working cycle. The time at which the combustion starts is determined by the timing of the ignition. If combustion be started too early the maximum cylinder pressure is developed before reaching top dead center, which results in a decrease in power output due to the piston working against

this pressure. On the other hand, if combustion be started too late the maximum cylinder pressure is not developed until the piston has travelled a considerable distance on the working

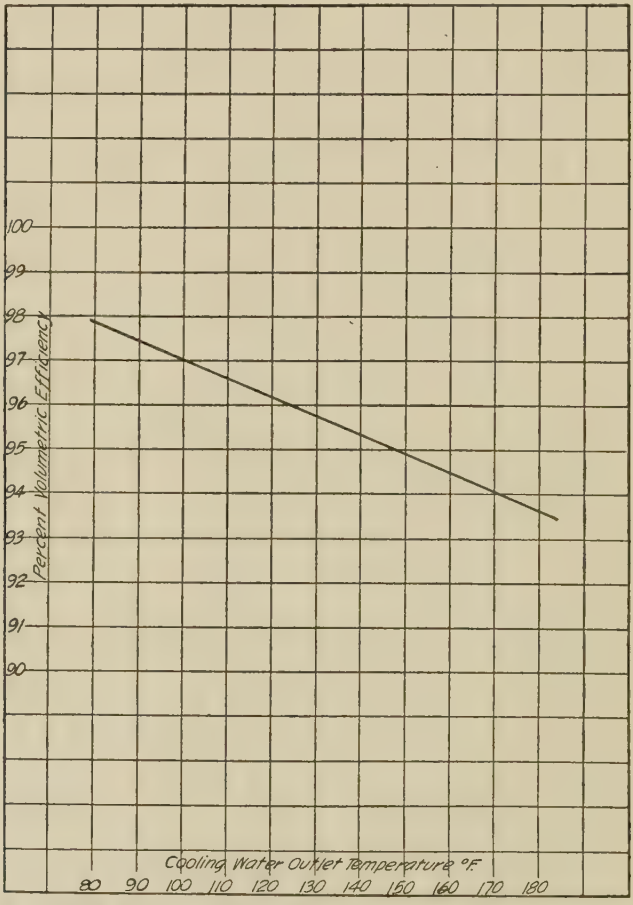


Figure 8. Effect of Cooling Water Temperature on Volumetric Efficiency

stroke. This also results in a loss in power due to failure to take advantage of the maximum pressure during the early portion of the expansion stroke. In general, the best time for ignition is that which gives the maximum power. The exact

timing which gives this condition depends largely on the speed of operation. Since the time required for combustion remains practically constant, the spark must be advanced more and more as the speed of the engine increases.

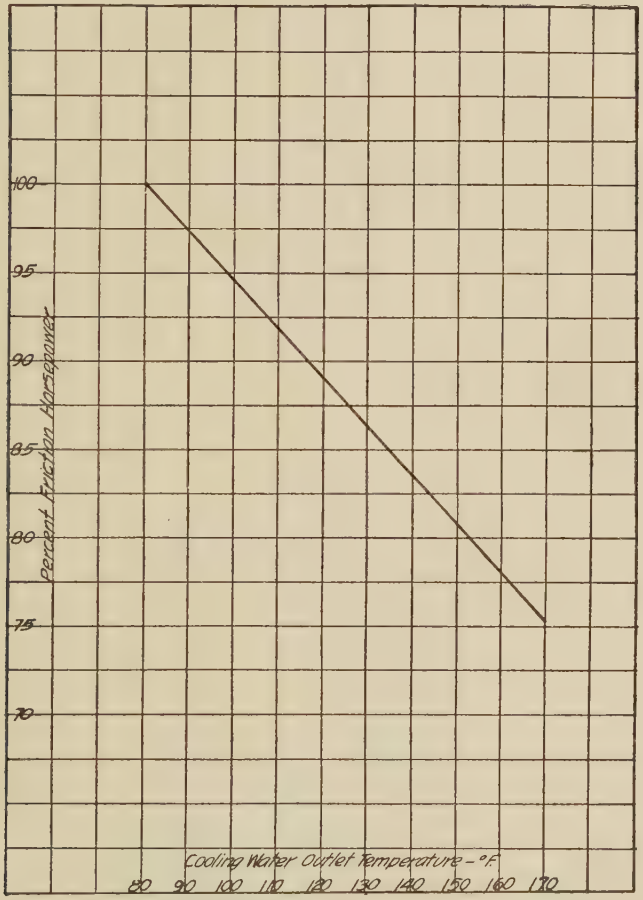


Figure 9. Effect of Cooling Water Temperature on Friction Horsepower

Figure 10 illustrates the effect of variation in spark timing upon the output of an engine. It will be noted that as the r.p.m. increases the advance required for maximum power

also increases. The adjustment of the ignition timing affects not only the power output but the efficiency of the engine as well. Improper timing of the ignition usually results in a

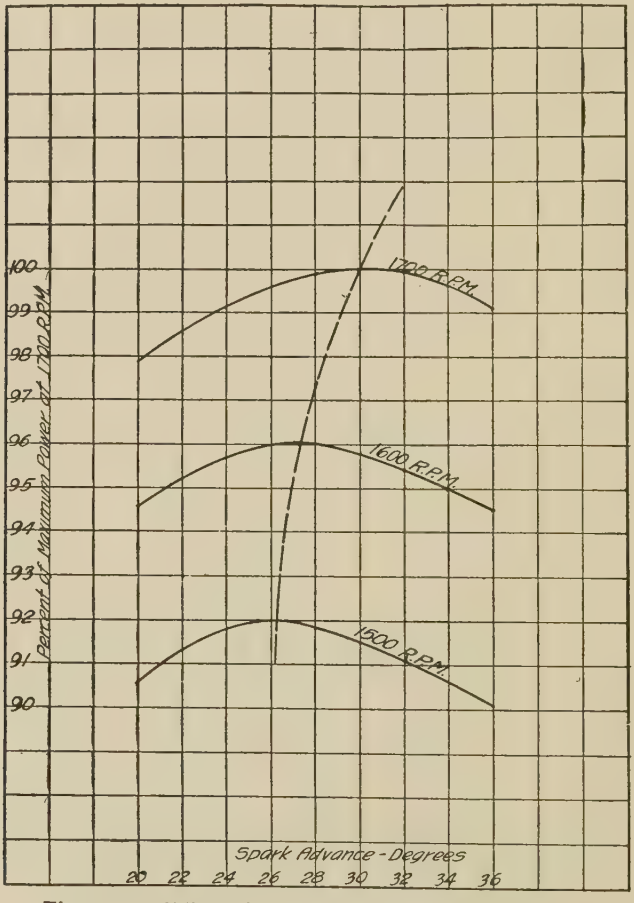


Figure 10. Effect of Spark Advance on Engine Power

tendency to overheat. An extremely early timing is apt to cause excessive cylinder pressures, resulting in severe strains on the engine and a tendency toward rough running.

Preignition. A condition similar to early timing of the spark frequently arises due to auto-ignition of the charge; that is, the charge is ignited before the spark takes place.

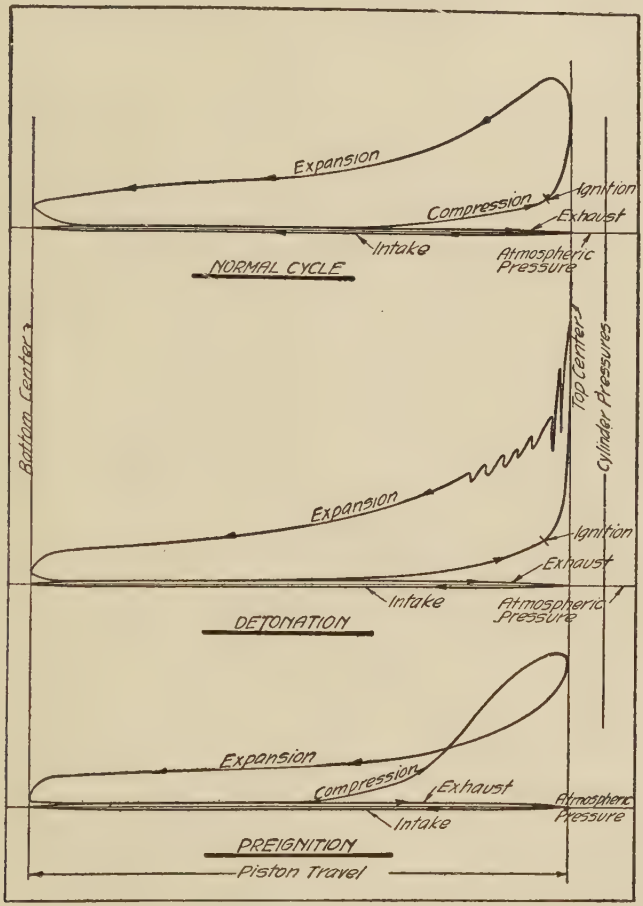


Figure 11. Typical Pressure-Volume Diagrams for Normal Cycle, Detonation, and Preignition

This condition is usually brought about by the presence of some particularly hot spot in the combustion chamber, such as hot exhaust valves, spark plug electrodes, or particles of in-

candescent carbon adhering to the piston or cylinder head. The tendency toward auto-ignition or preignition is increased by a higher compression ratio. During the compression stroke, the temperature of the charge increases as the pressure rises, hence the higher the compression the greater the tendency to self-ignition.

Diagrams in Figure 11 illustrate the effect of preignition on the working cycle. The upper diagram represents normal operation, while the lower diagram represents the condition prevailing when preignition is present. The charge is ignited early in the compression stroke and the pressure rises rapidly, reaching a maximum before dead center. This results in a considerable increase in the negative work during the compression stroke and also decreases the positive work during the expansion stroke, resulting in a considerable reduction in the power output. Preignition once started usually results in rapid overheating of the cylinder, thus becoming worse and worse as operation is continued, until further running becomes practically impossible.

Detonation. A phenomenon which is often associated with preignition, but which is of a distinctly different character, frequently occurs. This is known as detonation. Detonation consists in an entire modification of the nature of the combustion taking place in the engine cylinder. The character of the process is changed from that of an extremely rapid combustion to that of a violent explosion which results in a very sudden and intense rise in pressure. The conditions which bring about detonation are similar to those which cause preignition and the two phenomena frequently occur simultaneously.

The diagram at the middle of Figure 11 illustrates conditions in the cylinder when detonation is present. It will be noted that ignition starts in the normal manner but that the pressure rise resulting from combustion is much more rapid

and reaches a considerably higher value. The first violent explosion wave is frequently followed by secondary waves of less violence, which is indicated by the shape of the early part of the expansion curve. The principal factors in controlling detonation are the temperature and pressure at the end of the compression stroke and the character of the fuel used. In general, the higher the compression temperature or pressure, the greater the tendency toward detonation. The relation between the chemical composition of the fuel and the tendency toward detonation is not well understood, but there is a marked difference in the behavior of different fuels in this respect. Certain fuels, notably benzol and alcohol, are remarkably free from this difficulty and there are chemicals, commonly referred to as dopes, which, when added to ordinary gasoline, effectively suppress any tendency toward detonation.

As regards gasoline, the high-test gasolines are superior to low-test in this respect and the western and mid-continent products are superior to the eastern fuels.

With favorable conditions as regards cooling of exhaust valves, spark plugs and cylinder heads, domestic aviation gas may usually be used with compression ratios up to 5.5 to 1 without encountering serious trouble from detonation.

Compression ratio. Both the power output and economy of a gas engine are improved by an increase in the compression ratio. It should be noted that the compression ratio and not the compression pressure is the determining factor. In a gas engine the compression ratio and expansion ratio are equal, and since the useful power is derived from the difference between the work output during the expansion stroke and work input during the compression stroke, the further the expansion and compression are carried, the greater the amount of useful work which can be obtained from a given quantity of mixture. In addition to improving the power output and fuel economy, increasing the compression ratio raises the speed at which maxi-

mum power is developed. This effect may be attributed to the fact that with higher compression ratios the quantity of exhaust gas remaining in the cylinder at the beginning of the in-

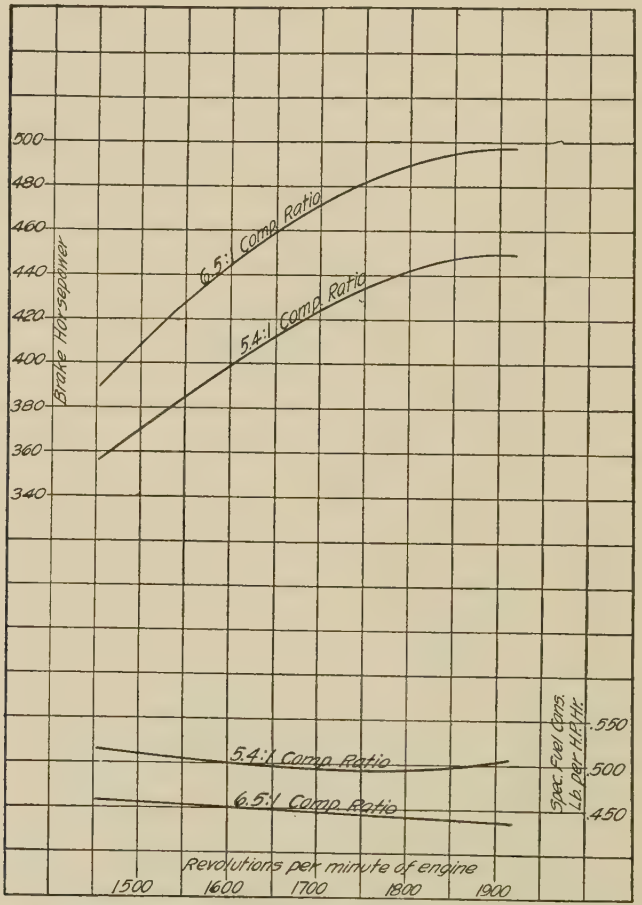


Figure 12. Effect of Change of Compression Ratio on the Performance of the Liberty Engine

duction stroke is reduced. This tends to improve the volumetric efficiency, thus raising the speed at which the power peak occurs.

Figure 12 gives a comparison of the power output and

fuel consumption of the Liberty engine at compression ratios of 5.4 and 6.5.

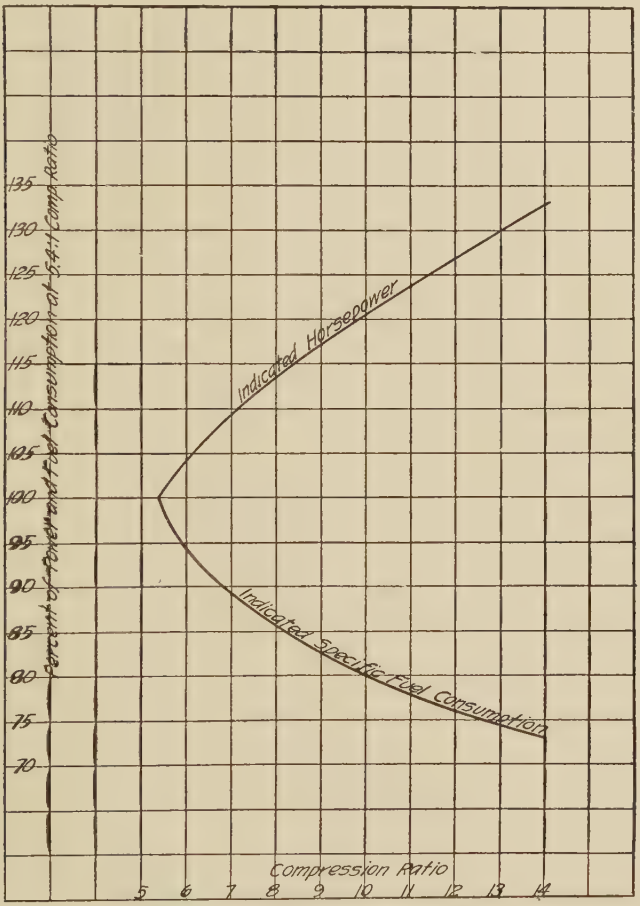


Figure 13. Variation of Indicated Horsepower and Indicated Specific Fuel Consumption with Compression Ratio

Figure 13 shows the variation of indicated horsepower and indicated specific fuel consumption with compression ratio. The values are expressed in per cent of the normal output and fuel consumption obtained at a compression ratio of 5.4.

Figure 14 gives a comparison between the theoretical and observed increase in efficiency resulting from an increase in

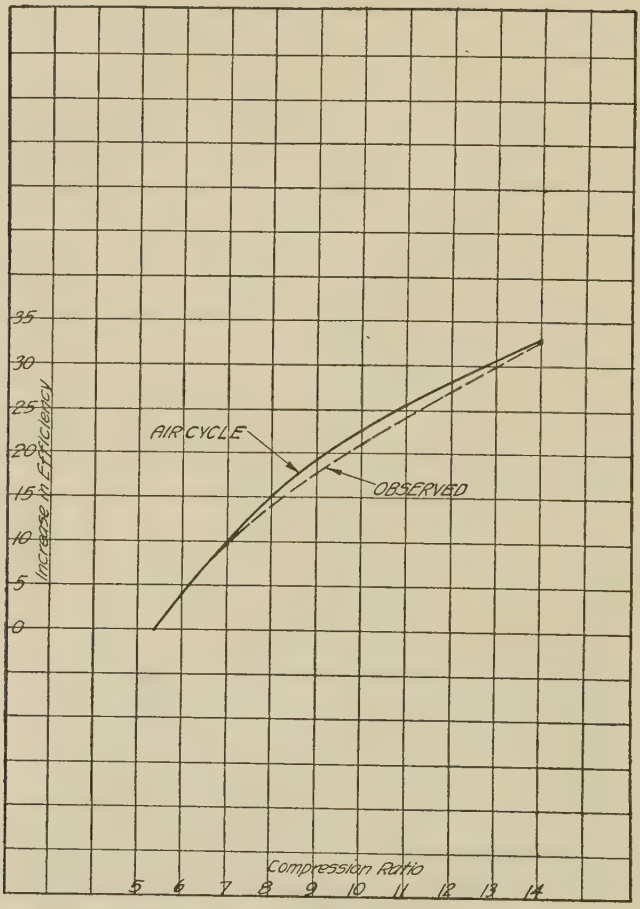


Figure 14. Theoretical and Observed Increase in Efficiency Resulting from Increase in Compression Ratio

compression ratio. The fuel used during these tests consisted of a mixture of benzol and aviation gasoline.

Since an increase in the compression ratio increases the tendency toward detonation and preignition, special fuels are

required for very high compression ratios. As stated above, a compression ratio of 5.5 to 1 is about the maximum which can be successfully used with ordinary aviation gasoline as fuel. For this reason the majority of aviation engines are designed to operate at approximately this ratio, since the necessity for special fuels largely offsets the advantage to be gained from the superior economy and power output obtained with high ratios. High compression ratios also have a very marked tendency to cause spark plug trouble.

Altitude. Since the density of the atmosphere decreases rapidly with altitude, the weight of charge drawn into the engine cylinder also falls off as the altitude is increased. This results in a rapid reduction in power output. As a first approximation the indicated horsepower should vary directly with the atmospheric density, and numerous tests conducted by the Bureau of Standards indicate that the actual variation follows this law closely when the temperature of the entering air is maintained constant.

The problem is complicated by the fact that the temperature also decreases rapidly as the altitude is increased. At very high altitudes the temperature becomes so low that the vaporization and distribution of the fuel is seriously interfered with and the drop in power is usually greater than that indicated by the considerations of density alone.

The friction loss of the engine does not fall off in proportion to the density, hence the mechanical efficiency becomes lower as the altitude is increased. For this reason the brake horsepower decreases more rapidly with increasing altitude than the indicated horsepower.

Figure 15 indicates the average results of a large number of tests conducted in the Altitude Laboratory of the Bureau of Standards. During these tests the temperature of the entering air was varied according to standard air tables. It will be noted that the indicated horsepower under these conditions

falls off more rapidly than the density, and that due to reduction in mechanical efficiency the brake horsepower falls off at a still greater rate.

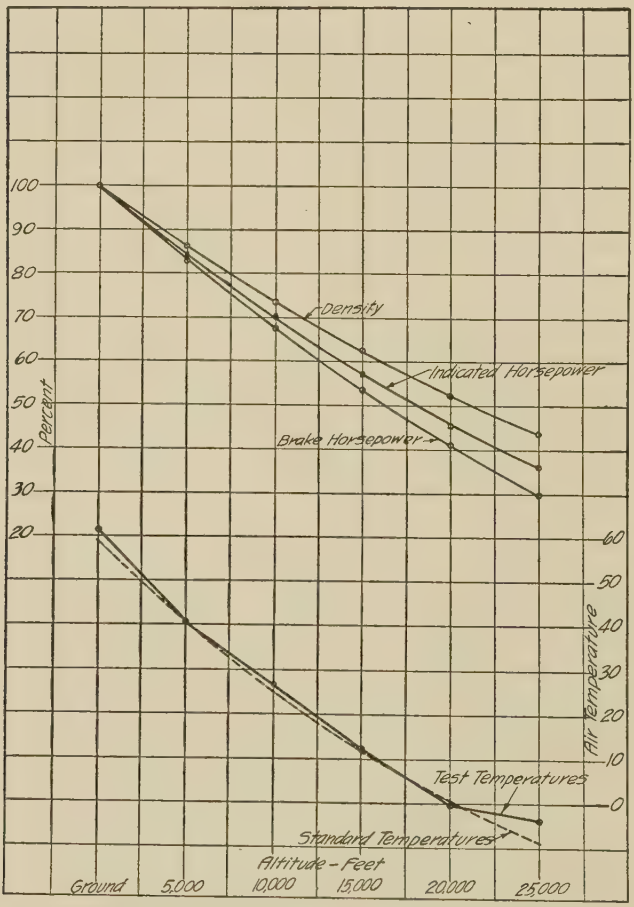


Figure 15. Average Results of Altitude Tests of Several Aircraft Engines

Figure 16 gives typical curves of horsepower against speed for various altitudes from sea level to 25,000 feet. Power outputs are expressed in per cent of normal sea level output. It will be noted that the peak of the power curve occurs at lower

speeds as the altitude is increased. This effect is very important if the particular engine being considered normally operates near its power peak at sea level. For under these

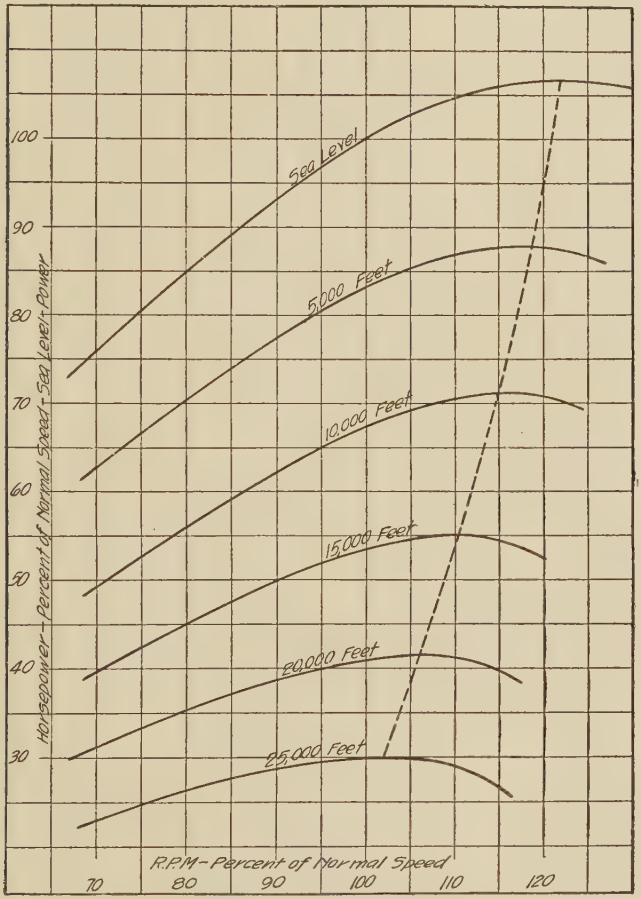


Figure 16 The Effect of Altitude on the Full-Throttle Performance of an Aircraft Engine

circumstances it is quite possible that at higher altitudes the engine would be operating at speeds above the power peak. This, of course, would result in a much more rapid decrease

in power output at increased altitude than would be the case with an engine operating well below its power peak. In this connection it should be noted that, as stated above, high com-

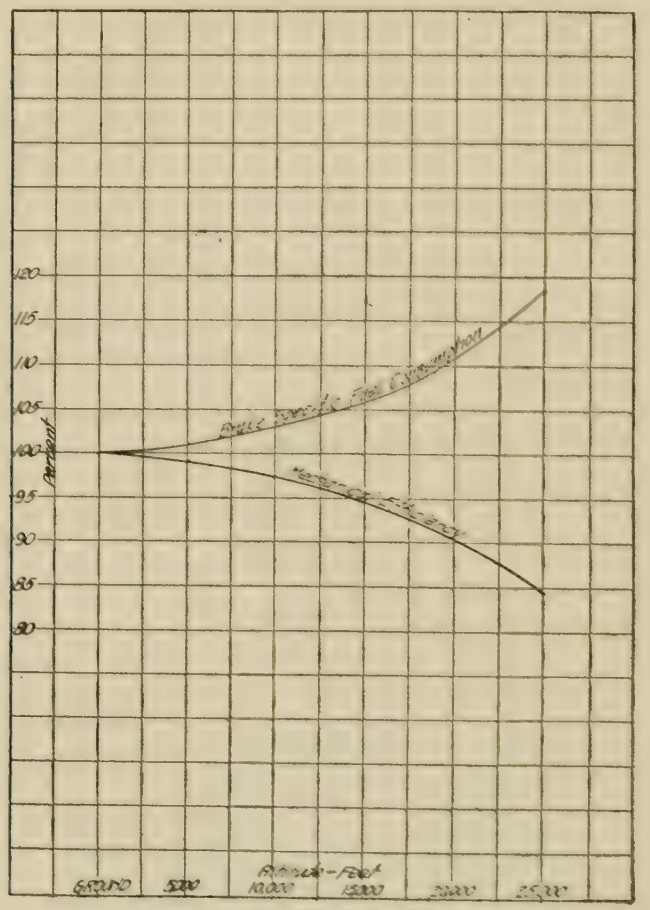


Figure 17. Effect of Altitude on Brake Specific Fuel Consumption and Mechanical Efficiency

pression ratios tend to raise the power peak. This effect may be important at high altitudes. Engines which normally operate near their power peak will have their output at high altitude increased by high compression due to this cause. In

this class of engines the beneficial effect of high compression may be much greater at altitude than at sea level. On the other hand, engines which are well below their power peak throughout the normal range of operation do not show this effect. Thus high compression may have a very beneficial effect on the altitude performance of engines operating near their peak. Whereas in engines operating well below their peak the per cent gained due to high compression is practically independent of altitude.

The efficiency of conversion of heat energy into power appears to be little affected by change of density with altitude. Therefore, if proper fuel mixtures are maintained, the indicated specific fuel consumption remains practically constant. However, due to reduction in mechanical efficiency, the brake specific fuel consumption increases with altitude very nearly as the reciprocal of the mechanical efficiency. This relation is shown in Figure 17. It must not be forgotten that these fuel consumption values hold true only when proper fuel mixtures are maintained. It is possible that at extremely high altitudes, with some types of induction system, the low atmospheric temperature may reduce the efficiency of heat conversion and so adversely affect the indicated specific fuel consumption. With the range of altitudes explored in recent altitude chamber tests, however, (0 to 25,000 feet), such an effect has not been apparent.

CHAPTER III

ENGINE TYPES

Definition of Types

Air-cooled and water-cooled. The most general classification of internal combustion engines for aircraft use is based on the method of cooling. Both air-cooled and water-cooled types may be further classified on the basis of cylinder and crankshaft arrangement. While the water-cooled engine has predominated in the past, recent developments have clearly indicated the possibility of building successful air-cooled engines of any power output now required, and this type deserves careful consideration. Comparing the air-cooled engine with the water-cooled, the former possesses the following advantages: reduction in weight due to elimination of water-cooling system; ease of installation and maintenance due to elimination of water-cooling system; elimination of danger of freezing or failure of water-cooling system through leaks; more rapid warming up of the engine before flight, and reduced vulnerability from projectiles. The chief advantage of the water-cooled engine over the air-cooled engine appears to lie in the greater latitude possible in the design of the engine and the arrangement of the cowl, supports, housing, etc., forming a part of its installation. There being no necessity for keeping in mind the provision of sufficient air flow, the engine and its installation may be arranged in the manner most convenient, whereas, with the air-cooled engine, certain desirable arrangements cannot be employed because they interfere with the flow of cooling air to and from the engine cylinders.

Cylinder arrangements. Engines are commonly classified according to their cylinder arrangement. The National Ad-

visory Committee's Report on Nomenclature for Aeronautics defines certain of these types. On the other hand, there are distinct types which are not covered by this report. The following definitions are given merely as an aid to the discussion which follows:

Vertical engine. An engine having its cylinders vertical above the crankshaft.

V or Vee engine. An engine having its cylinders arranged in two rows, forming in the end view the letter "V."

W engine. An engine having its cylinders arranged in three rows forming in the end view the letter "W." (Sometimes called the "Broad-Arrow type.")

Inverted engine. An engine having its cylinders below the crankshaft.

X *engine.* An engine having four rows of cylinders whose axes meet at the crankshaft center in a manner resembling the letter "X."

Flat engine. An engine having cylinders arranged in two rows on opposite sides of the crankshaft.

Radial engine. An engine having stationary cylinders arranged radially around a common crankshaft. An engine having cylinders disposed radially to the crankshaft center in two planes is commonly referred to as a two-row radial.

Rotary engine. An engine having revolving cylinders arranged radially around a common fixed crankshaft.

Barrel-type engine. An engine having its cylinders arranged equidistant from and parallel to the main shaft.

Comparison of Types

The selection of the most suitable type of engine for any particular service must be based upon considerations of power output, torque variation, speed, balance, convenience of arrangement and cooling.

The smaller engines may be operated satisfactorily with fairly poor torque characteristics and without being in dynamic balance. On the other hand, large engines require excellent dynamic balance and low torque variation.

Assuming uniform firing order, the torque variation is dependent upon the number of cylinders; the larger the num-

ber, the smoother the torque curve. For a given power output the larger the number of cylinders the lighter the reciprocating parts, which makes higher crankshaft speeds possible. The selection of a suitable number of cylinders can not, however, be made on a basis of torque and cylinder size only, the question of dynamic balance must also be considered as certain combinations are necessary from this point of view.

A discussion of the question of engine balance may be found in any standard work on engine design. Judge's "Automobile and Aircraft Engines" contains an excellent discussion of balance characteristics of the types most frequently used for aircraft engines. Six-cylinder-in-line engines and eight-cylinder-in-line engines may be balanced without the use of balance weights by properly arranging the crankshaft throws. Multiples of these arrangements, such as twelve-cylinder Vees, eighteen-cylinder W's or sixteen-cylinder Vees are, of course, also in balance. Single row radials may be balanced by counter-weights attached to the crankshaft. Double or triple row radials may also be balanced by the same method. The four-cylinder-in-line engine is not in dynamic balance and can not be put in balance by balance weights rotating at crankshaft speed. The eight-cylinder Vee, combining two banks of four cylinders each, is not in balance as usually constructed. However, by a special arrangement of the crankshaft throws and the use of counter-weights on the crankshaft, the engine may be made to balance properly.

Water-cooled types. For water-cooled engines having outputs between 250 and 1,000 horsepower, the twelve-cylinder 60° V appears to possess the majority of advantages. The general arrangement is convenient from the point of view of installation and maintenance. The cylinder sizes are reasonable and the reciprocating parts may be made light enough to permit operation at fairly high speed. The torque characteristics are excellent and the dynamic balance is satisfactory.

For water-cooled engines below 250 horsepower the six-in-line and the eight-cylinder 90° V are both suitable; the six-in-line being superior as regards simplicity and dynamic balance, while the eight cylinder possesses the advantage with regard to torque variation, compactness and weight.

For engines over 1,000 horsepower the W type has much to recommend it. The cylinder size may be kept within reason; the torque reaction is very uniform; the balance is satisfactory, and the overall dimensions are smaller than could be obtained with a twelve. The reciprocating parts being lighter than those of the twelve, the eighteen-cylinder W could be operated at somewhat higher speed. For a given displacement, the weight would probably be in favor of the 18 as compared to the 12.

The radial engine and the X engine appear to have little to recommend them in water-cooled types.

Air-cooled types. Considering the air-cooled type, the cylinder size and cylinder arrangement have a very pronounced bearing on the cooling possibilities. The cylinders must be so located that an air blast of considerable velocity can be provided, and in general, the larger the cylinder the more difficult is the cooling.

Considering cooling problems alone, the radial engine probably offers the best arrangement. Single row radials require an odd number of cylinders in order to secure uniform firing and have been built in three, five, seven, nine and eleven cylinders. All of these types may be satisfactorily balanced and the selection of the number of cylinders depends mainly on cylinder dimensions and torque variation. The three-cylinder type is useful only in very small sizes, while nine cylinders have proved to be sufficient to give satisfactory torque characteristics up to 500 horsepower. For outputs in excess of 500 horsepower, the two-row radial, the twelve-cylinder V or some form of X engine appears to be best suited.

While the majority of air-cooled engines have been of the radial type, considerable success has recently been attained in the development of the air-cooled Vee engine in sizes up to 400 horsepower. Cooling of this type is effected by bringing air down the V of the engine and allowing it to pass out at the side between the engine cylinders. Comparing this type with the nine-cylinder radial, the reduction in blast is partially offset by the fact that having twelve cylinders, the cylinders are smaller and easier to cool. The torque curve of the twelve-cylinder engine is smoother and the projected area of the engine is reduced.

Description of Representative Engines

A table giving the principal characteristics of representative American aircraft engines will be found at the end of this chapter. Space does not permit a detailed description of each of these types. The descriptions which follow are considered typical of the various classes of engines now in use.

Packard 1A-1551 engine. The Packard 1A-1551 engine used in the U.S.S. "Shenandoah" is the only American engine developed primarily for airship use. It is a six-cylinder vertical, water-cooled engine with 6⅝ inch bore and 7½ inch stroke. The displacement is 1,551.24 cubic inches and the compression ratio 6½:1. The general construction of the engine is shown by Figures 18 and 19 and its installation dimensions by Figures 20 and 21. The cylinders are individual, built up of steel forgings and with sheet steel water jackets welded on. The crankcase is box section type parted horizontally at the crankshaft center line. The crankshaft is a single forging with seven main bearings, drilled and plugged for positive lubrication to bearings and connecting rods. The connecting rods are H-section with oil tubes to conduct oil to the piston pins and cylinder walls. The pistons are aluminum

die castings with three rings above and one below the piston pin. Two intake and two exhaust valves are used in each cylinder. They are actuated through yoked rocker arms by push rods at either side of the engine. Two camshafts are used. The lubrication system is dry sump with triple gear oil pumps. The oil strainer is in duplicate to permit cleaning during operation and includes in the same unit the high and

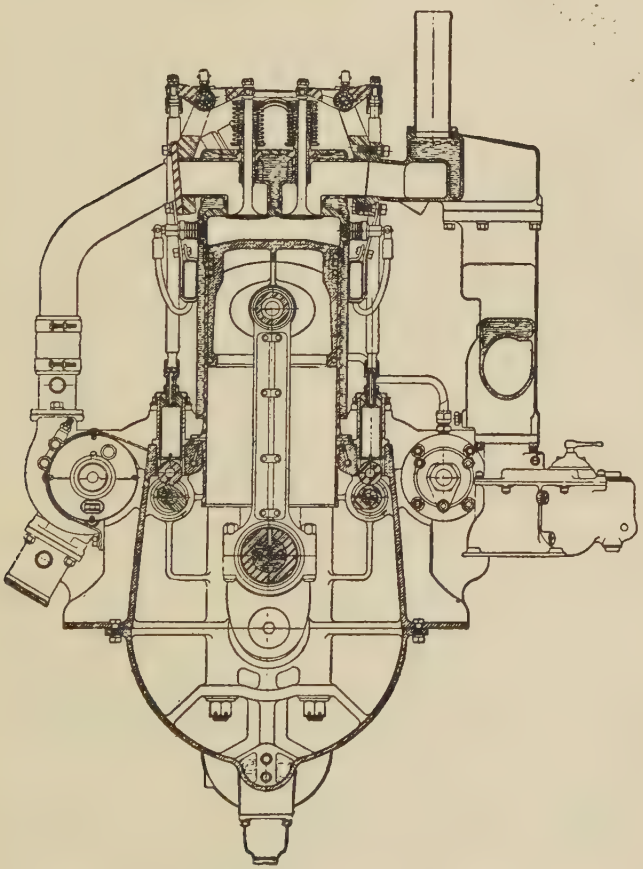


Figure 18. Packard 1A-1551 Engine Transverse Section

low pressure relief valves. The carburetor is a Stromberg NA-ZD5 with back suction mixture control. Ignition is sup-

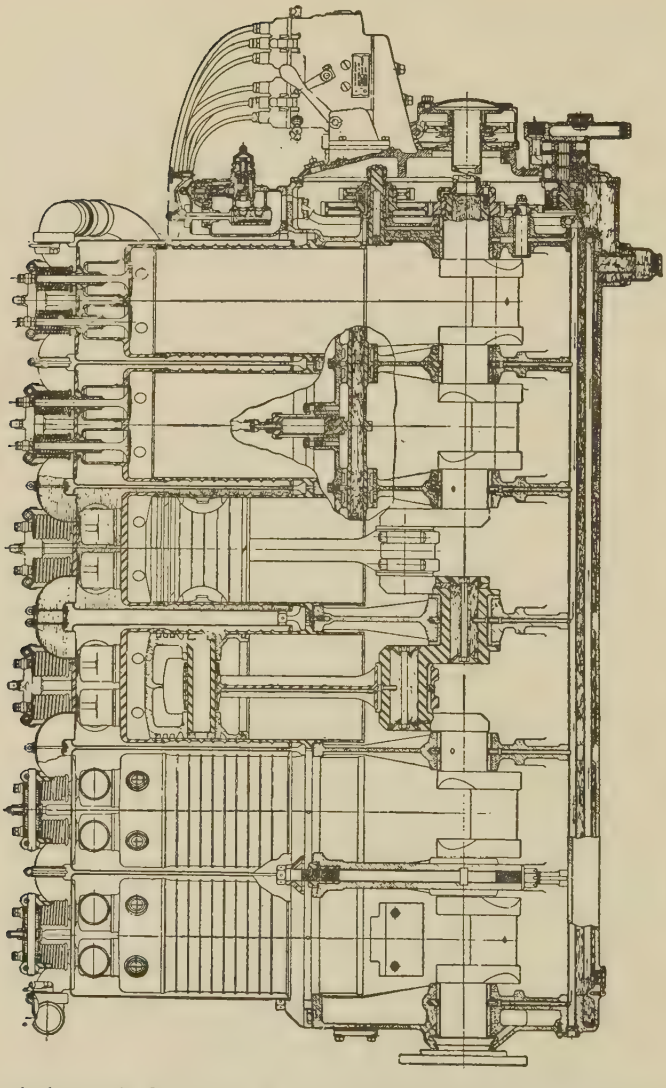
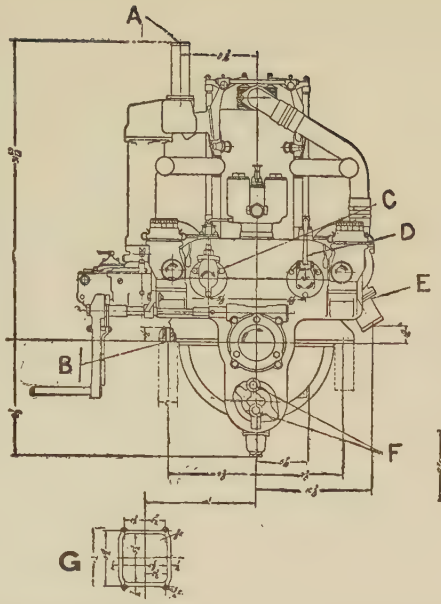


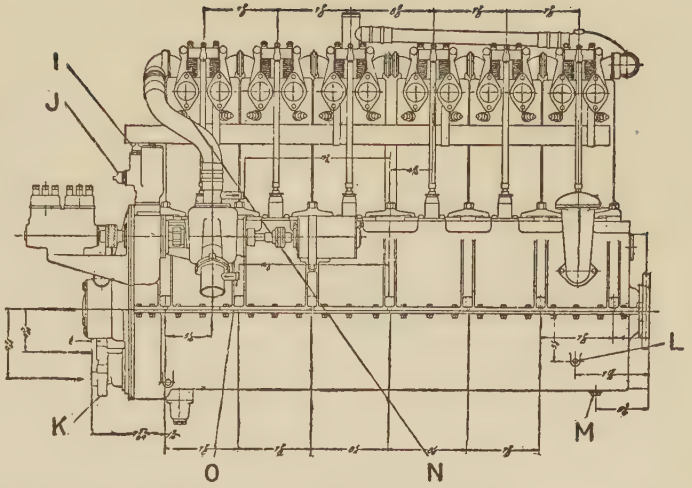
Figure 19. Packard 1A-1551 Engine Longitudinal Section

plied through four spark plugs per cylinder by two Delco six-cylinder duplex systems. A 12-volt constant current type generator runs at 1½ crankshaft speed. A centrifugal type governor acts on the throttle through oil from the high-pressure system and operates to hold the engine speed below 1,500



A 1⅞ inside diameter hose connections water outlet
B Drill 13/32 14 holes
C Tachometer drive A. S. U. S. A. standard ½ engine speed
D Compression release lever
E 2½ inside diameter hose connection water pump inlet
F ¾ pipe threads
G View looking up at carburetor air intake

Figure 20. Packard 1A-1551 Engine Installation Diagram, Rear End View



I Oil low pressure adjustment
J Oil high pressure adjustment
K Oil thermometer boss
L ⅜ pipe tap oil tank vent connection
M Oil drain
N ¾ inside diameter hose connection to starter engine water outlet
O ¾ inside diameter hose connection to starter engine water inlet

Figure 21. Packard 1A-1551 Engine Installation Diagram, Right Side

r.p.m. and to stop the engine when the oil pressure falls below 25 pounds per square inch. The engine is started by a hand crank with 10 to 1 gear ratio.

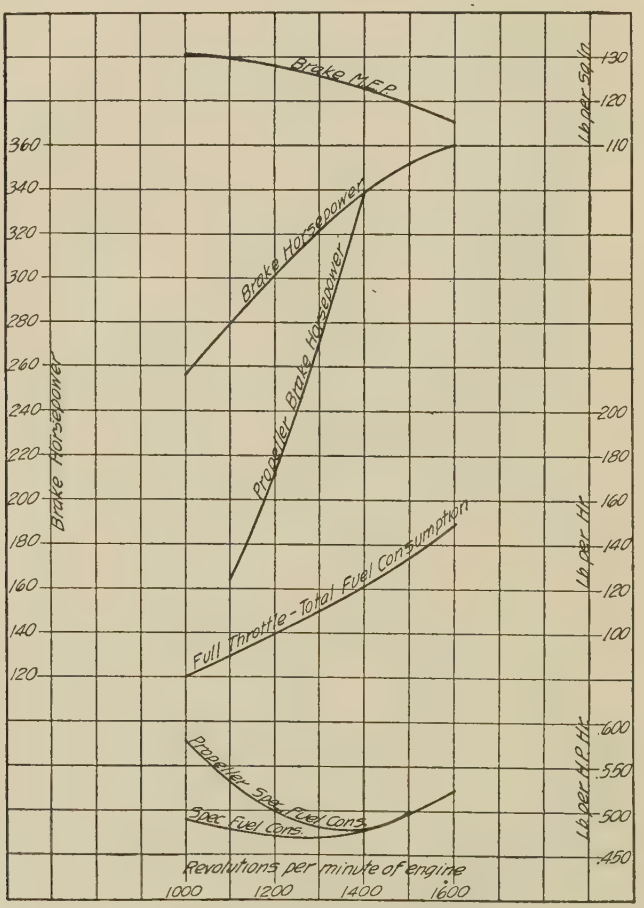


Figure 22. Packard 1A-1551 Engine Performance Curves

The following performance data are quoted by the manufacturer. Test results obtained at the Bureau of Standards are shown in Figure 22:

Maximum brake horsepower at 1,400 r.p.m	350
Normal rated brake horsepower at 1,400 r.p.m	300
Maximum fuel consumption at 1,400 r.p.m. normal rated horsepower	.50 lb./B.hp./hr.
Normal fuel consumption at 1,400 r.p.m. normal rated horsepower	.45 lb./B.hp./hr.
Maximum oil consumption at 1,400 r.p.m. rated hp	.03 lb./B.hp./hr.
Normal oil consumption at 1,400 r.p.m. rated hp	.02 lb./B.hp./hr.
Weight of engine complete including instrument board and hand starter	1,100 lb.
Weight of water in engine	50 lb.

Curtiss D-12 engine. The Curtiss D-12 engine (Figures 25 and 26) is a twelve-cylinder water-cooled 60° Vee engine with 4½-inch bore and 6-inch stroke. The total piston displacement is 1,159 cubic inches. The general construction of the engine is shown by Figures 23 and 24. The crankcase is of cast-aluminum alloy and consists of three sections, the crankcase proper on which the cylinders and main bearings are located, the oil pan covering the lower half of the crankcase and the gear case which is bolted to the crankcase end and carries the accessory drive gear train. The cylinders are of built-up steel and aluminum construction. Forged steel closed end barrels are screwed into aluminum head castings. The water jackets are formed by a separate aluminum casting which is attached by studs to the head casting and surrounds the entire group of six barrels. The lower ends are sealed by rubber rings under a flange on the barrels, compressed when the jacket casting is drawn up. The valve seats are machined directly in the steel ends of the barrels. The crankshaft is a single steel forging with eight plain babbitt-lined main bearings and one single row ball thrust bearing. The propeller hub is carried on tapers on the crankshaft end and located by a key. The crankshaft is made hollow; the main bearing journal bores are plugged to retain oil and the crankpin bores are fitted with tubes placed diagonally to carry oil to the crankpin bearings. The accessory drive gear is carried on splines in the inside of the rear main bearing journal. The connecting rods are steel

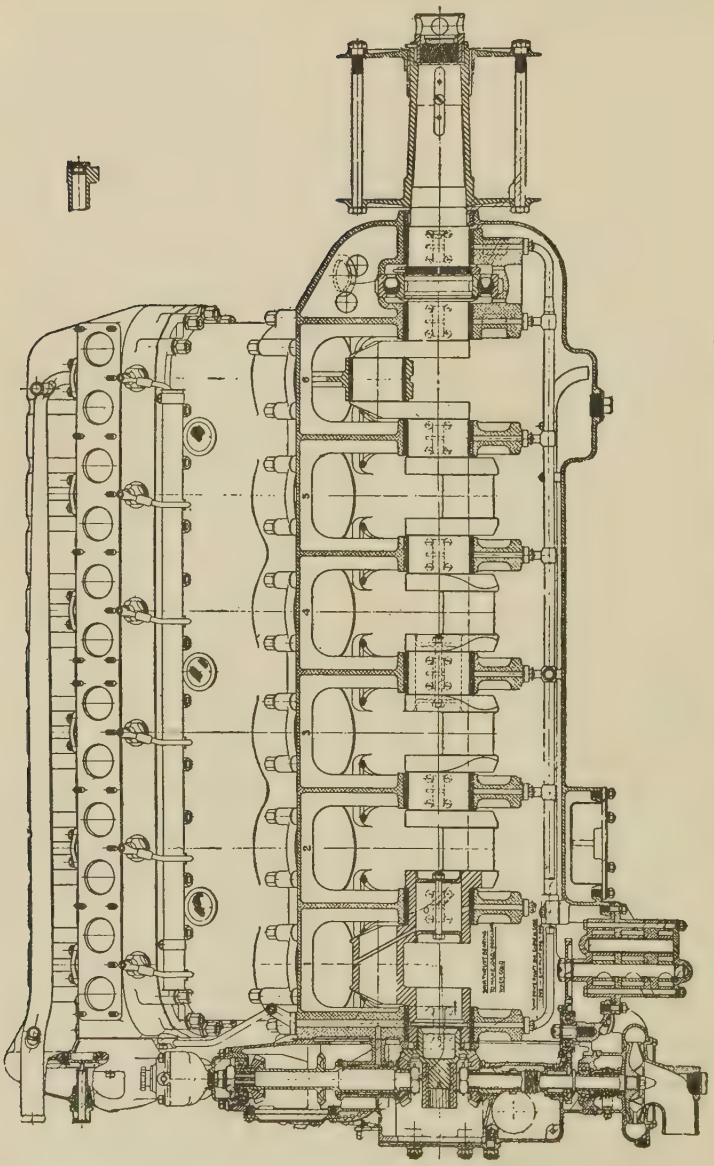


Figure 23. Curtiss D-12 Engine Longitudinal Section

forgings of "H" section. The secondary rod is carried on a link pin in a boss at the side of the main rod big end. The piston and link pin bearings are bronze bushings carried in the rods. The pistons are short straight cylindrical pistons of aluminum alloy ribbed across the inside of the heads. Three

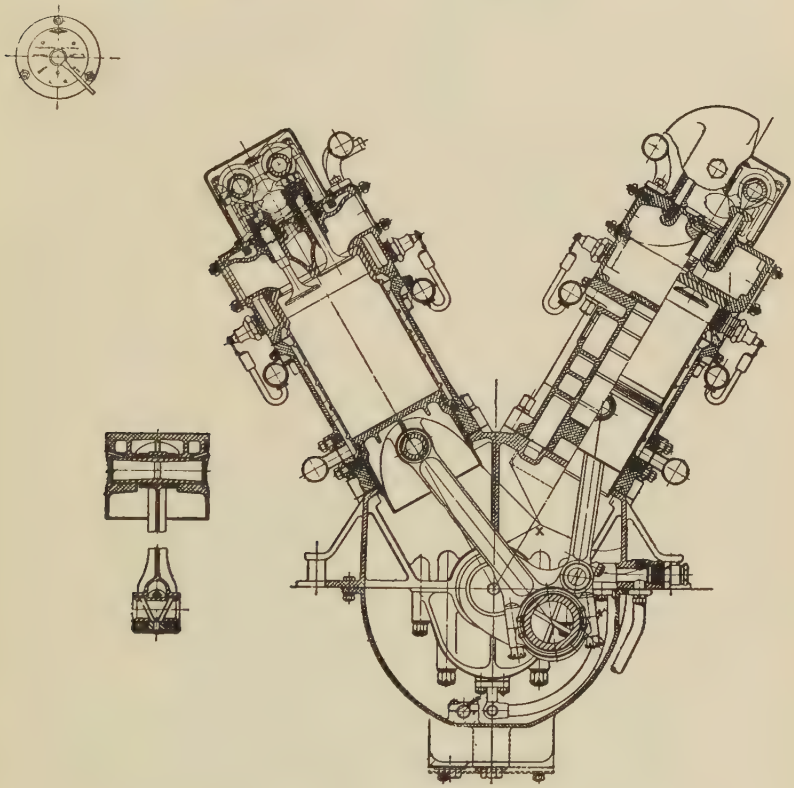


Figure 24. Curtiss D-12 Engine Transverse Section

narrow rings are fitted above the piston pin, but none below. The piston pin floats in the piston and rod upper end; it is retained in place in the piston pin bosses by snap rings. Two intake and two exhaust valves are used in each cylinder operated by two camshafts extending along the top of each

block. Pairs of valves in each cylinder are operated by single cams through yoked followers located in guides between the valves. Silchrome tulip valves are used. The lubrication is dry sump with pressure to the main bearings, rod bearings and camshaft bearings and by splash to the cylinder walls and piston pins. Oil is supplied by a single gear pump at 100 pounds per square inch pressure. Two gear scavenge pumps are located in a removable unit with the pressure pump at the bottom of the oil pan. The carburetor is a Stromberg NA-Y5D with back suction type of mixture control. Ignition is supplied by two Splitdorf SS-12 magnetos through two spark plugs in each cylinder.

The engine was designed for normal operation at 2,000 r.p.m. but is now run at a normal speed of 2,250 r.p.m. at which speed it develops 440 horsepower with a compression ratio of 5.3 : 1. The brake mean effective pressure at that output is 133.5 pounds per square inch. The weight of the engine is 690 pounds, or 1.57 pounds per horsepower.

Air-cooled Liberty engine. The air-cooled Liberty engine (Figures 27 and 28) is a Vee type air-cooled engine built by substituting air-cooled cylinders for the water-cooled cylinders of the Liberty engine. It differs from the water-cooled Liberty only in the cylinder, piston, induction system, and valve gear assemblies. The crankcase, crankshaft, connecting rods and main accessory drive train, are identical with those of the water-cooled engine. In making the engine air-cooled, the bore was reduced to 4⅝ inches thus providing cooling space between cylinders and reducing the piston displacement to 1,411 cubic inches. The crankcase is of cast-aluminum in two sections parted at the crankshaft centerline, the upper half carrying the cylinders and the main bearing upper halves; the lower half supporting the main bearing lower halves. The crankshaft is a single steel forging drilled and plugged for oil passages. The propeller hub is mounted on a

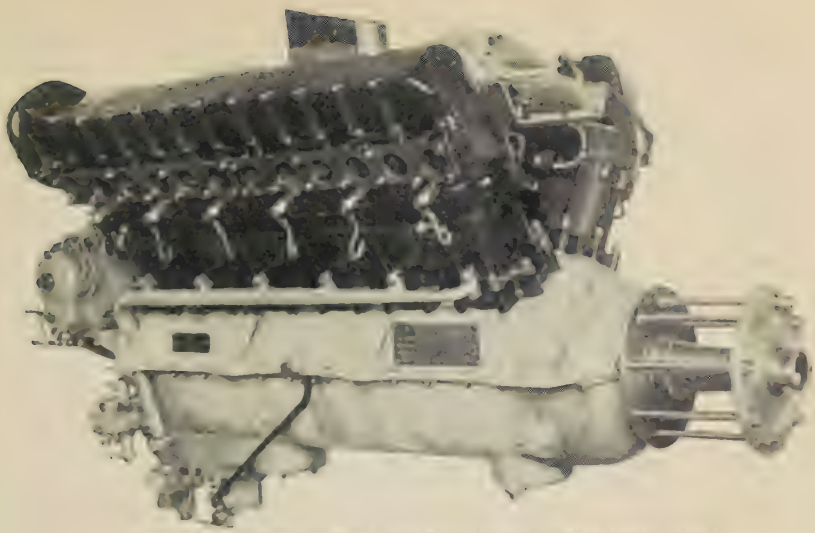


Figure 25. Curtiss D-12 Engine (three-quarter front view)

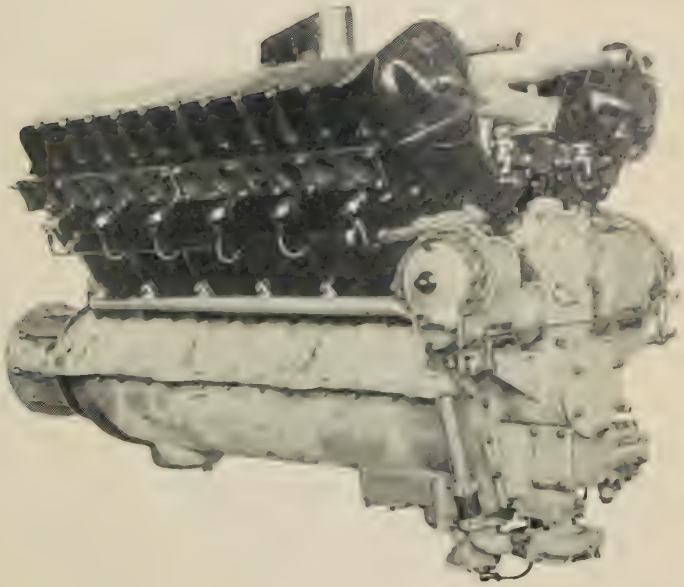


Figure 26. Curtiss D-12 Engine (three-quarter rear view)

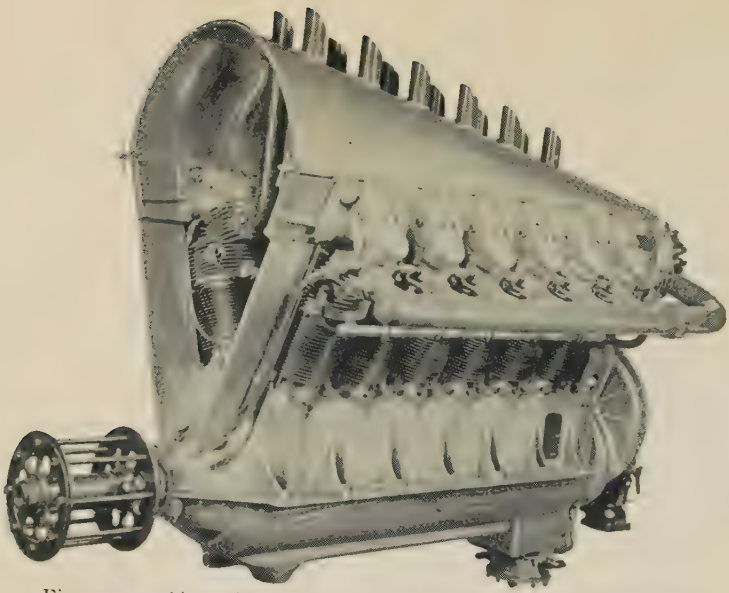


Figure 27. Air-cooled Liberty Engine (three-quarter front view)

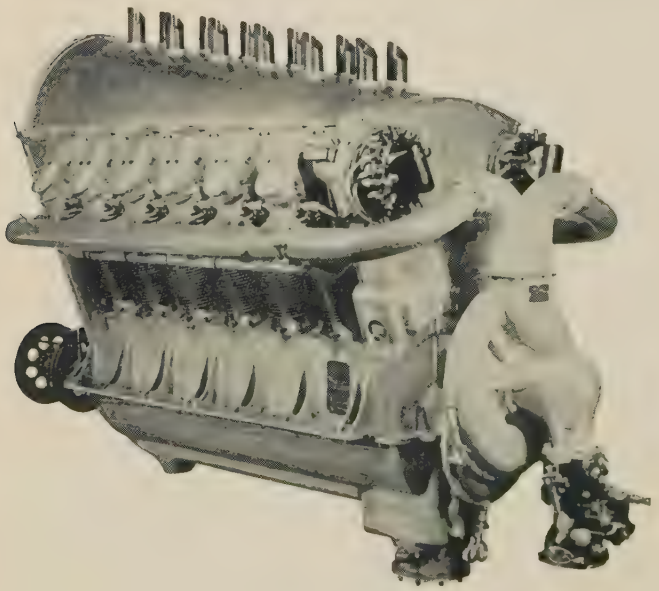


Figure 28. Air-cooled Liberty Engine (three-quarter rear view)

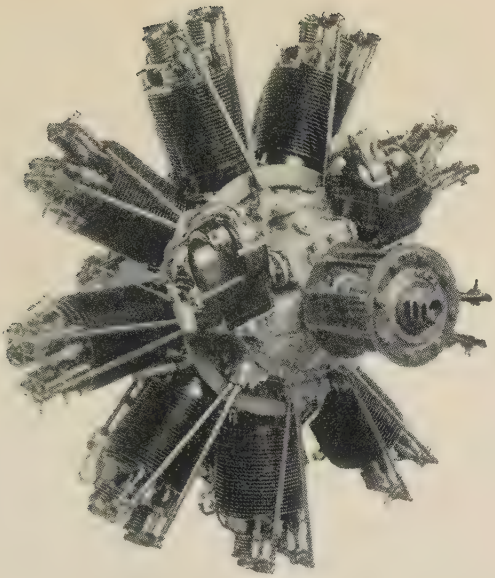


Figure 29. Wright J-4 Engine (three-quarter front view)

Figure 30. Wright J-4 Engine (three-quarter rear view)

taper and located by a key. The main accessory drive gear is bolted to a flange at the rear end of the shaft. The connecting rods are of "H" section, the secondary rod mounted on the center of the main rod bearing and working directly upon the bronze shell of the bearing. The pistons are cylindrical with three rings above the piston pin and one in the skirt. The cylinders have steel barrels with integral fins screwed and shrunk in a cast aluminum-alloy head. For additional security a steel ring is shrunk on the aluminum head below the lowest fin clamping the head to the barrel. The combustion chamber top is hemispherical with bronze valve seats shrunk in. The valves are tulip shaped; the exhaust valve is cooled internally by a partial filling of sodium and potassium nitrate. One intake and one exhaust valve are used in each cylinder. They are actuated by rocker arms from a single camshaft on each bank. The valve mechanism is fully enclosed and runs in a bath of oil. The fuel mixture for the engine is supplied by a single Stromberg NA-S8J carburetor through a gear driven low altitude supercharger at the rear of the engine. Ignition is by the standard Liberty-Delco system.

The engine has been built in upright and inverted types and with both geared and direct drive. With a compression ratio of 5.4 : 1 it develops 436 horsepower at 1,900 r.p.m. The brake mean effective pressure at that output is 128 pounds per square inch. The weight of the ungeared engine is 1,010 pounds, (2.32 pounds per horsepower), nearly 200 pounds lighter than the standard Liberty engine with radiator and cooling water.

Wright J-4 engine. The Wright J-4 engine (Figures 29 and 30) is a nine-cylinder air-cooled radial engine of 4½-inch bore and 5½-inch stroke. The crankcase is of cast-aluminum alloy divided vertically into three sections, the valve gear chamber, crankcase proper and induction chamber. The crankshaft is an integral steel forging with a single counterweighted

throw. The connecting rod group consists of a master rod with eight auxiliary rods attached to its big end by link pins. All of the connecting rod loads, therefore, are carried on the master rod big end bearing. The master rod itself is of the "H" section type, while the link rods are of the tubular type. The pistons are plain cylindrical pistons of aluminum alloy. The cylinders consist of a cast aluminum head and jacket liberally finned for heat dissipation. A steel sleeve is screwed into this jacket, which sleeve is provided with a hold-down flange for attaching the cylinder to the crankcase. The valves are silchrome, tulip type operated by exposed push rods and rocker arms. Lubrication is by pressure, to connecting rod bearings, cams and cam followers. The oil is supplied by a gear pressure pump and removed from the sump by a single large capacity scavenge pump of the gear type. The rocker arms are fitted with Alemite connections for lubrication. Two Splitdorf SS-9 magnetos supply ignition through two plugs in each cylinder. One Stromberg NA-U5 carburetor supplies the fuel mixture through three circular induction passages.

The engine develops 210 horsepower at 1,800 r.p.m. (a brake mean effective pressure of 117.4 pounds per square inch) with 5.1 : 1 compression ratio. The weight is 480 pounds giving a weight-horsepower ratio of 2.28 pounds per horsepower.

Engine Data

A complete tabulation of the characteristics of aircraft engines would include much data of little value and thus become unnecessarily bulky. The following table has been restricted to engines of American design and manufacture. An effort has been made to cover the complete range of powers available without including data on engines which are now considered obsolete.

Characteristics of American Aircraft Engines

Engine Name and Model	Normal B. hp.	Normal r.p.m.	Cylinder Arrangement	No. of Cyl.	Bore (inches)	Stroke (inches)	Displacement (cu. in.)	Compression Ratio	Weight Dry (lb.)	Weight per B. hp. (lb.)	Normal *BMEP* (lb./sq. in.)
Curtiss D-12	440	2,250	60° Vee	12	4.500	6.000	1,159.0	5.3	690	1.57	133.0
Curtiss V-1400	510	2,100	60° Vee	12	4.875	6.250	1,420.0	5.5	680	1.33	135.4
Liberty 12A	420	1,700	45° Vee	12	5.000	7.000	1,648.9	5.4	875	2.08	118.6
Liberty Air-cooled (Inverted)	436	1,900	45° Vee	12	4.625	7.000	1,411.0	5.4	1,010	2.32	128.5
Packard 1A-1500	510	2,100	60° Vee	12	5.375	5.500	1,498.0	5.5	735	1.44	128.4
Packard 1A-1551	337	1,400	Line	6	6.625	7.500	1,551.0	6.5	1,060	3.15	122.9
Packard 1A-2500	800	2,000	60° Vee	12	6.375	6.500	2,489.6	5.8	1,130	1.41	126.7
Wright E-4	204	1,800	90° Vee	8	4.724	5.118	718.0	5.35	474	2.32	125.0
Wright J-4	210	1,800	Radial	9	4.500	5.500	787.0	5.1	480	2.28	117.4
Wright L-4	60	1,800	Radial	3	4.250	5.250	223.5	5.0	175	2.92	118.1
Wright T-3	525	1,800	60° Vee	12	5.750	6.250	1,947.0	5.3	1,160	2.21	118.6
Pratt and Whitney "Wasp"	400	1,900	Radial	9	5.750	5.750	1,344.0	5.25	675	1.69	124.0

CHAPTER IV

ACCESSORIES

Ignition

Of the two general types of aircraft engine ignition—battery and magneto—neither seems to have any considerable superiority. The decision favoring one or the other must be influenced largely by the character of the remaining equipment of the aircraft under consideration, for both battery and magneto ignition systems can be made very reliable and rather light in weight. Their advantages and disadvantages as listed below seem to be fairly well balanced.

Battery ignition.

Advantages:

1. Considerable advantage from the point of view of weight when batteries are required also for other equipment.
2. Requires no "booster" for starting.
3. Operates satisfactorily with retarded spark.
4. Voltage is not affected considerably by engine speed.
5. Subject to less mechanical trouble in drives and working parts because of lightness of moving parts.
6. More freedom in mounting breaker mechanism and distributor for accessibility.

Disadvantages:

1. Fire risk in case of accident because of "live" wires.
2. Somewhat more complicated than magneto system.
3. More wiring to be shielded to prevent radio interference.
4. Generally heavier when other electrical equipment is not needed.

Magneto ignition.

Advantages:

1. Simplicity for maintenance. Of considerable importance where servicing is contemplated during flight.
2. Lighter than battery system when other electrical equipment is not required.
3. Shorter wiring for radio shielding.
4. Electrical fire risk stops when engine stops.
5. Less likely to be affected by accidents to other electrical equipment.

Disadvantages:

1. Requires "booster" for starting unless fairly rapid turning of the engine is possible.
2. Voltage increases with speed.
3. Often adversely affected by retarding spark because of movement away from maximum flux density.
4. Requires heavier drives and contains more moving parts.
5. More limited as to location because of higher unit weight and heavier drives.

Starters

Aircraft engine starters designed to be installed as an engine accessory and carried with it are of five general types: compressed air, gas, electric, hand gear and inertia starter.

The *compressed air starter* consists of a small compressed air motor supplied with air from containers carried in the aircraft. The motor is so placed that it can be connected through suitable gears to the engine crankshaft for starting. The compressed air starter has the advantage that it can be so designed that it requires no "kick-back" disconnecting device. Its disadvantages are that it requires the use of a relatively heavy air container, an air pump for replenishing the air supply and

is capable of only a rather small number of engine revolutions if the weight of the container is held within reasonable limits.

Gas starters consist ordinarily of some device to pump an explosive mixture into the engine cylinders with sufficient pressure to cause the crankshaft to rotate. This type of starter has the general advantage that it supplies to the engine a more readily combustible mixture than is frequently obtainable from the carburetor. The gas may be supplied from a pressure tank in which case the system has the disadvantages of the air starter with the added fire risk and impossibility of replenishing the supply or it may be supplied as is done in the case of the Bristol starter by a small two-cylinder engine, one cylinder of which is a power cylinder and the other a pumping cylinder. The Bristol type starter necessitates hand starting of the small engine. The gas starter also needs no "kick-back" safety provision but does require a gas distributor.

The *electric starter* consists simply of a small electric motor with automatic engagement through dogs or splines to the engine crankshaft driving through considerable reduction gearing. A disengaging device is provided to prevent injury to the gears or drive in case of "kick-back." The electric starter requires the use of a rather heavy battery and its extent of utility is limited by the capacity of the battery.

The *geared hand starter* consists simply of a hand crank which may be connected through worm gearing to the crankshaft. A disengaging device must be provided to prevent injury to the operator. The hand starter has the very attractive advantage of being capable of unlimited engine turns but is of necessity geared so high that the highest possible engine speed is often rather too slow for effective starts.

The *inertia starter* is also a hand starter but is provided with a small flywheel and clutch so that by turning the hand crank the flywheel may be made to rotate at very high speed. The clutch is then engaged and the crankshaft turned by the inertia of the flywheel. The inertia starter also is capable of

unlimited starts, requires no source of power and turns the engine shaft rather rapidly for several turns without serious exertion. Its only disadvantage appears to be the possibility of mechanical troubles resulting from the high speed of the parts and the rather severe loading during engagement. It compares favorably with the best of other starters in weight.

Carburetors

There are only two types of aircraft carburetors in general use in the United States, the Zenith and Stromberg, differing essentially in their manner of load compensation as indicated by the discharge well types and in the idling system arrangement. Those differences will be considered in more detail in the following paragraphs.

The float chamber with the float mechanism and the needle valve serves to maintain constant fuel level at the metering jets and discharge passages. The Stromberg and Zenith carburetors differ in the type of float and needle valve linkage employed; the Zenith carburetor having a circular float with a concentric needle valve and weighted levers as shown in Figure 31, while the Stromberg employs the single lever with the float attached to one end as shown in Figure 32.

Main fuel system. The Zenith carburetor employs a double main fuel system to compensate for natural enrichment with increasing load. Referring to Figure 31, the discharge nozzle will be seen to consist of a main discharge nozzle extending up through the center with the compensating discharge nozzle surrounding it. The main discharge nozzle is fed from the fuel passage through the main metering jet at its lower end. The mixture from the main discharge nozzle tends to become richer with increased load. The compensating discharge nozzle is fed from the fuel passage through the compensating jet and the secondary well which is vented to the atmosphere. The compensating mixture therefore becomes leaner as the load is increased and the two metering jets are so proportioned

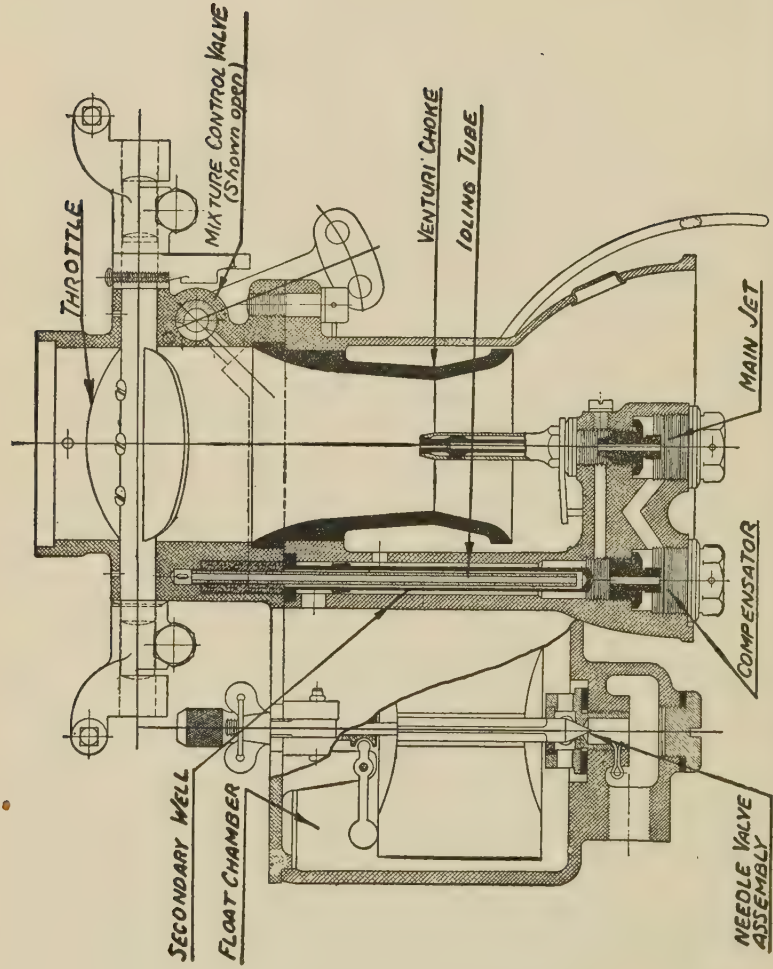


Figure 31. Sectional View of 12-Cylinder Liberty Engine Zenith U. S.-52 Carburetor Assembly

that the mixture is approximately constant throughout the range.

The Stromberg carburetor employs the air bleed principle of compensation for load enrichment. Figure 33 shows the general arrangement of the Stromberg carburetor. All the

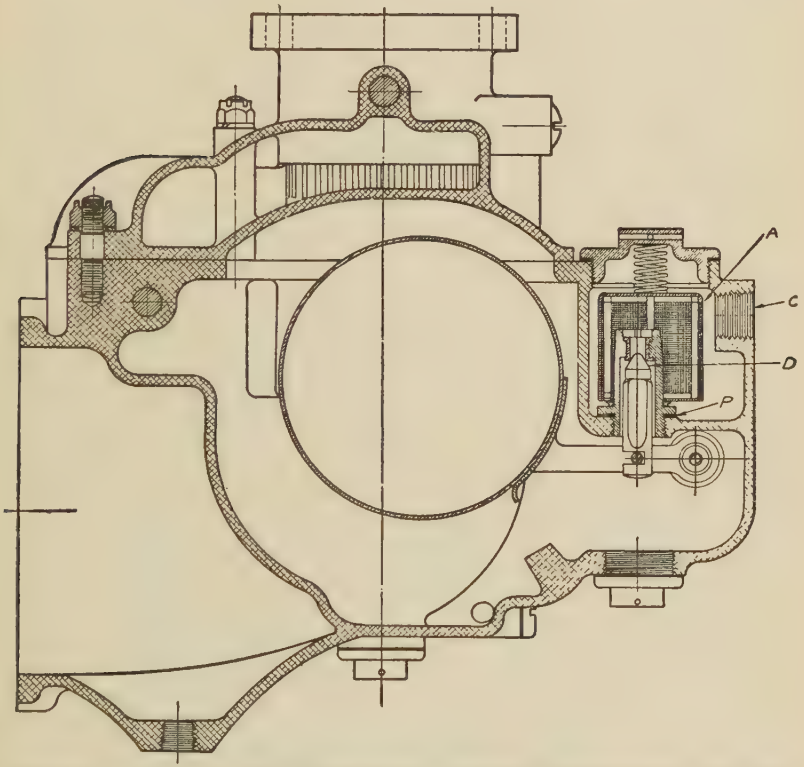


Figure 32. Sectional View of Stromberg Carburetor Showing Float Chamber and Float Mechanism

fuel flow in the carburetor is controlled by the fuel metering jet placed either at the bottom of the main discharge tube or in the main fuel passage from the float chamber. The holes drilled in the wall of the main discharge tube communicate with the accelerating well, the idle passages and the air bleed

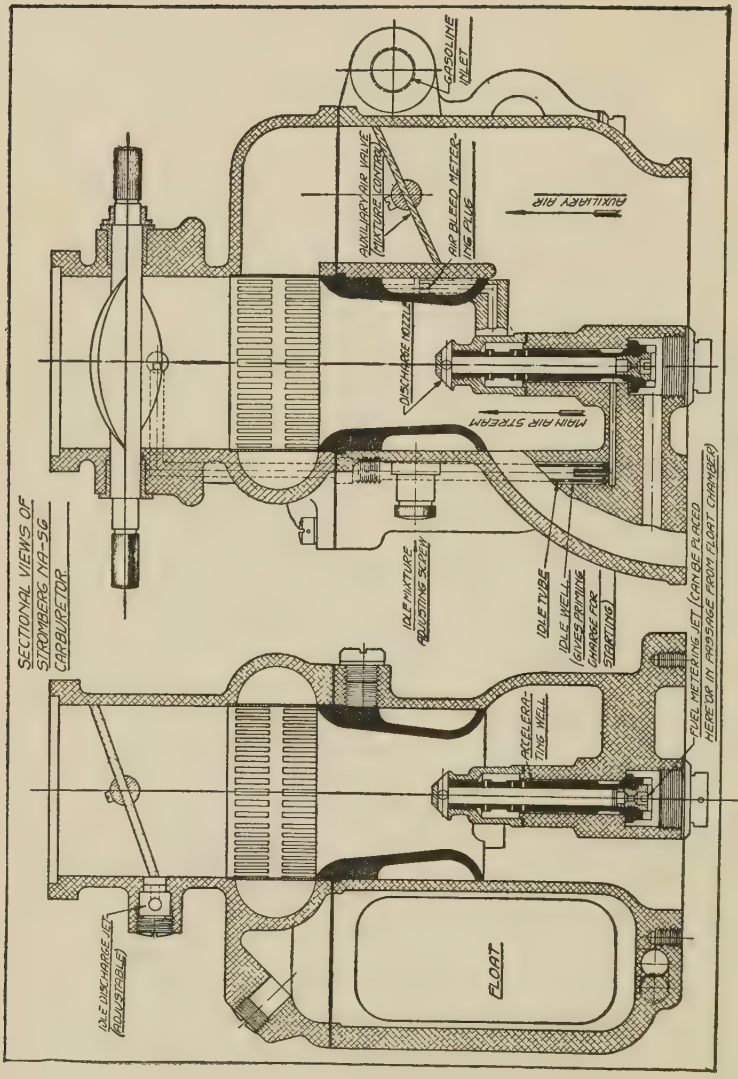

SECTIONAL VIEWS OF STROMBERG NA-S6 CARBURETOR
GASOLINE INLET
AUXILIARY AIR VALVE (MIXTURE CONTROL)
AIR BLEED METERING PLUG
AUXILIARY AIR
DISCHARGE NOZZLE
MAIN AIR STREAM
IDLE MIXTURE ADJUSTING SCREW
IDLE TUBE
IDLE WELL (GIVES RICHNESS CHARGE FOR STARTING)
ACCELERATING WELL
FUEL METERING JET (CAN BE RAISED HERE OR IN PASSAGE FROM FLOAT CHAMBER)
IDLE DISCHARGE JET (ADJUSTABLE)
FLOAT

Figure 33. Sectional Views of Stromberg NA-S6 Carburetor

metering plug. Compensation is effected by the increase of air drawn in through the holes in the main discharge tube wall as the load increases. The compensation is adjusted by variation in the size of the hole in the air bleed metering plug.

Acceleration. The rapid discharge of fuel required for acceleration with sudden opening of the throttle is provided in the Stromberg carburetor by the accelerating well surrounding the main discharge tube and in the Zenith by the secondary well surrounding the idling tube. The fuel for acceleration accumulates in the Stromberg by flow through the compensating holes in the discharge tube during the light load operation. In the Zenith the accelerating charge is supplied through the compensating jet and accumulates in the secondary well.

Idling system. The general schemes of idling in the two carburetors are largely similar. The sources of supply of fuel however are different. The idling fuel in the Stromberg is supplied through the main metering jet and accelerating well. In the Zenith the idling fuel enters the secondary well through the compensating jet. Idling is adjusted in the Zenith by variation of the size of hole in the bottom of the secondary well. The emulsion for idling is drawn up through the idling tube and discharged through the hole shown above the throttle. The idling mixture in the Stromberg carburetor is adjusted by variation of the supply of air to the idle well by manipulation of the idle adjusting screw.

Mixture control. Several methods are employed for the adjustment of mixture strength to compensate for normal variation due to altitude or other causes. The commonest of them, known as the "back-suction" control, reduces fuel flow by reducing the pressure in the float chamber. The reduction of pressure is accomplished in two ways, both by communication with the carburetor barrel, but with different valve locations. The "suction bleed back-suction" control has a fixed orifice vent in the float chamber and a variable opening be-

tween the float chamber and the carburetor barrel. Then opening the valve reduces the float chamber pressure and therefore reduces fuel flow. The "air bleed back-suction" control has the fixed orifice between the float chamber and the carburetor barrel and the variable opening as a float chamber vent. Closing the valve in this case reduces the pressure and fuel flow. The first, the "suction bleed" control, is used on the Zenith carburetor for the Liberty engine, shown on Figure 31. The second, the "air bleed" control, is used on the Stromberg NAZ-D5 carburetor for the Packard 1551 engine.

A second type of control reduces the depression at the discharge nozzle. Two principal methods of accomplishing this have been employed. One, the least prominent, moves the carburetor venturi so that the region of highest depression is removed from the discharge nozzle. The second admits air in considerable quantity between the venturi and the throttle and so reduces the venturi depression. Such a control is illustrated in Figure 33.

A third general type of mixture control reduces the fuel flow between the float chamber and the nozzle either by bleeding air or imposing some adjustable mechanical restriction such as a needle valve.

CHAPTER V

FUELS AND LUBRICATION

Fuels

Aircraft engine fuel. As previously stated, the gas engine may be operated on any fuel sufficiently volatile to be successfully introduced into the cylinder by mixing with the ingoing air. However, many other characteristics must be considered in determining the suitability of a fuel for use in aircraft engines. Among these may be mentioned: cost, quantity available, facility of storage and transportation, heating value, combustion characteristics and tendency toward detonation, freezing point, chemical stability, chemical activity or tendency to attack metals, chemical composition and chemical activity of products of combustion, freedom from solid residue, and tendency to absorb moisture from the atmosphere. While either liquid or gaseous fuel may be used satisfactorily in the gas engine, considerations of transportation and handling almost necessitate the use of liquid fuel for aircraft. While there are a number of liquid fuels which might possibly be used, gasoline appears to best satisfy the majority of the requirements mentioned above, and is the only fuel which has been extensively used for aircraft engines.

Gasoline. Gasoline manufactured from crude petroleum possesses many advantages for aircraft use. The supply is abundant and the price reasonable; the heating value is high compared to other liquid fuels available; when properly distilled it is free from tendency to combine chemically with materials used in the fuel system; its volatility is satisfactory; there are no solid or gummy residues resulting from evaporation; the products of combustion are relatively non-poisonous,

and satisfactory combustion may be secured over moderately wide ranges of the fuel-to-air ratio. The freezing point is well below any temperature likely to be encountered.

Crude petroleum consists of a mixture of a large number of compounds which are members of the various hydro-carbon series. Each of these series consists of an indefinite number of compounds of carbon and hydrogen, the proportions and manner of combination following a definite law. For instance, the paraffin series is made up of compounds whose chemical formulae take the form $C_nH_{(2n+2)}$. Starting with the simplest compound we have CH_4, C_2H_6, C_3H_8 . . . . . $C_nH_{(2n+2)}$. Other important series in the chemistry of petroleum are Napthene series represented by the general formulae C_nH_{2n} and the Aromatics series represented by $C_nH_{(2n-6)}$. As the number of atoms composing a molecule is increased, the specific gravity and boiling point are raised. Crude petroleum not only contains a great number of compounds in any one series but also contains members of several series. The total number of distinct chemical compounds in a given sample of crude is very large. Their separation and identification is practically impossible. The crude may be readily analyzed to determine the percentage of carbon and hydrogen, but this furnishes no information as to the manner in which these compounds are combined and hence throws little light upon the physical properties of the sample.

If crude oil be subjected to heat, the lighter and more volatile portions start to distill off at rather low temperatures, and as these distill off the boiling point increases until all volatile matter has passed off and impurities only remain. During the latter portion of such distillation, after the boiling temperature passes 750°F., the distillation is usually accompanied by a chemical breaking down of the compounds known as cracking. This consists in splitting up the heavier and more complicated molecules into simpler molecules accompanied by the formation of free carbon. Those portions of the crude which

distill off at relatively low temperatures are grouped together under the general name of gasolines. This substance has no fixed chemical composition; hence, its physical properties vary

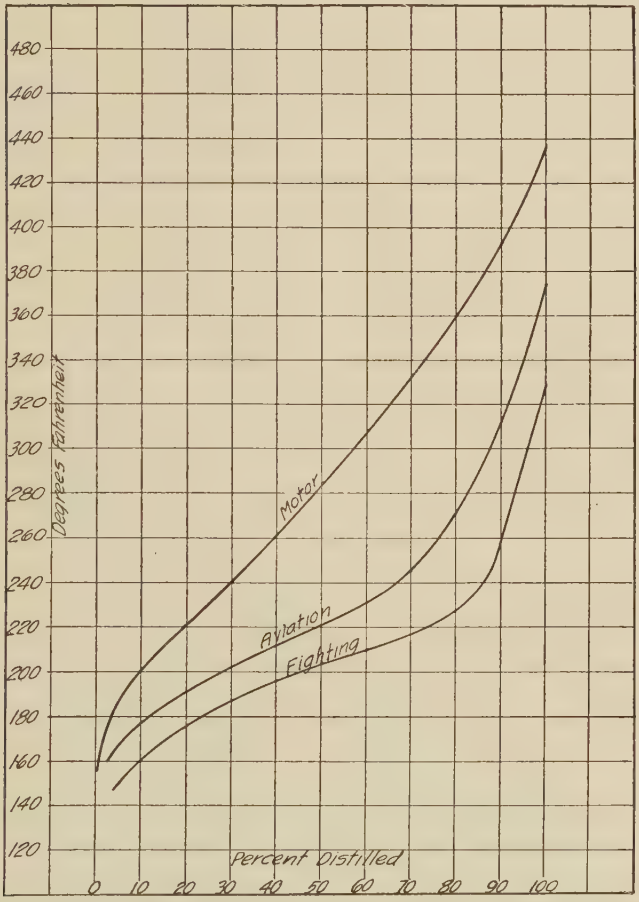


Figure 34. Typical Distillation Curves of Standard Aviation Gasoline

considerably and there is no generally accepted standard of volatility. In general, all compounds which are suitable for use as motor fuels are considered gasolines.

The distillates of the next group, which are not sufficiently volatile to be used as gasoline, are the lightest burning oils commonly called kerosene. It is customary to specify a narrower range of boiling points and a higher volatility for aviation gasolines than is required for automobile service. This is illustrated in Figure 34 which gives the distillation curves of standard aviation gasoline, specially volatile gasoline, and ordinary motor gasoline. It will be noted that while the motor gasoline contains some elements quite as volatile as those in the aviation gasoline, the percentage of high boiling elements is greater and the maximum boiling point is much higher. Reference should be made to current Specifications of the Army and Navy Air Services for further details as to the requirements of suitable aviation gasoline. These include specific gravity, freedom from acidity, and freedom from water and other impurities.

Since the chemical composition of aviation gasoline is not definitely fixed, the heating value of the fuel varies somewhat according to its composition. However, this variation is not wide and the heating value is usually not far from 20,000 B.t.u. per pound. This is considerably in excess of the heating value of most common fuels and exceeds that of any of the non-petroleum liquid fuels which are available for use as substitutes for gasoline. It has been found that the chemical composition of the gasoline varies considerably with the source of the crude from which it was manufactured; this variation in chemical composition produces a variation in the tendency of the fuel to induce detonation when operating in the engine. Crudes which contain considerable quantities of elements from the paraffin series appear to be the least desirable in this respect. Gasolines from the California crudes, being low in paraffin content have by far the least tendency toward detonation, while the Pennsylvania crudes, being high in paraffin content, are the worst. The Mid-Continent crudes are intermediate between the Pennsylvania and California crudes in this respect.

Alcohol. Alcohol has frequently been considered as a suitable fuel for aviation engines; its chief advantages being uniform chemical composition, high volatility, low freezing point

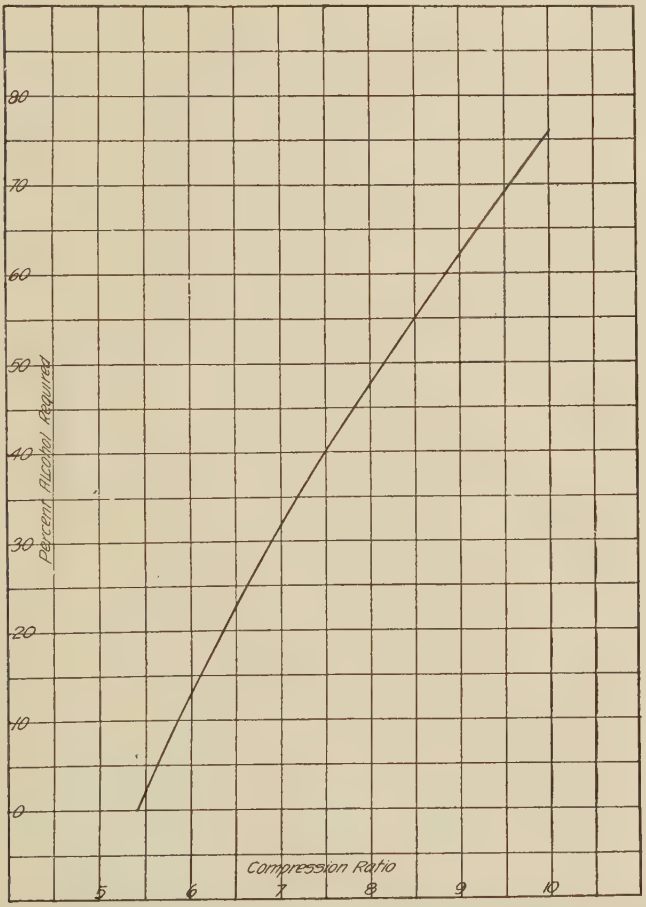


Figure 35. Percentage of Alcohol Required in Alcohol Aviation Gasoline Fuel Mixtures to Prevent Detonation With Various Compression Ratios

and marked anti-detonating properties. Its chief disadvantages are low heat of combustion, tendency to absorb moisture, and for the present at least, scarcity and high price. Its

low heat of combustion constitutes the most serious disadvantage for aircraft use, the value being about 12,000 B.t.u. per pound, which is only slightly in excess of half that of gasoline. The total weight of fuel required for a given power output is therefore considerably higher, which appears to render its use prohibitive as aviation fuel. Anti-detonating qualities of alcohol render it possible to utilize it in engines having higher compression, thus raising the efficiency and partially off-setting the effect of its low heat of combustion. However, in spite of this effect the total weight of fuel required still remains much higher for alcohol than for gasoline. The anti-detonating properties of alcohol have been utilized by blending this fuel with gasoline. Relatively small percentages of alcohol blended with gasoline greatly decrease the tendency toward detonation and permit higher compression. Figure 35 represents the results of tests conducted by the Bureau of Standards to determine proportions of alcohol and gasoline required for various compression ratios. The chief difficulty connected with the use of alcohol to suppress detonation lies in the fact that alcohol blends with gasoline only when no moisture is present. Alcohol being highly hygroscopic tends to absorb moisture from the atmosphere and it is very difficult to keep alcohol or alcohol-gasoline blends entirely free from water. The following table gives the results of experiments conducted by the Engineering Division of the Air Service to determine the quantity of water required to produce separation in alcohol-gasoline blends of various proportions:

Alcohol-Gasoline Mixtures	Distilled Water Required to Produce Separation in 10 cc. of Fuel at Room Temp.	
*Denatured alcohol 100%	6	cc.
" " 50% plus 50% aviation gasoline	0.35	cc.
" " 30% " 70% " "	0.20	cc.
" " 15% " 85% " "	2	drops
" " 5% " 95% " "	1	drop

* Denatured in accordance with formula 28A (1% Aviation Gasoline).

Benzol. Benzol has been widely used as motor fuel and is, in many respects, suitable for aviation use. The anti-detonating properties are equal or superior to those of alcohol;

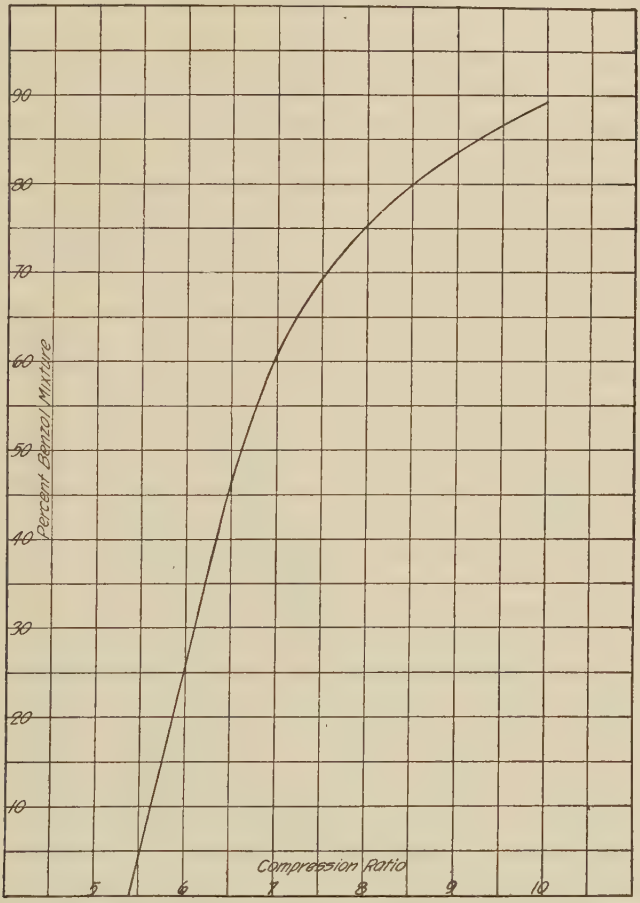


Figure 36. Percentage of Benzol Required in Benzol Aviation Gasoline Fuel Mixture to Suppress Detonation with Various Compression Ratios

its heat value, approximately 17,500 B.t.u. per pound, is nearly equal to that of gasoline and considerably above that of alcohol. The chief disadvantages are scarcity and high freezing point.

Blends of benzol and gasoline have been much used in aircraft engines due to the anti-detonating properties. Benzol blends readily with gasoline in all proportions and is not precipitated

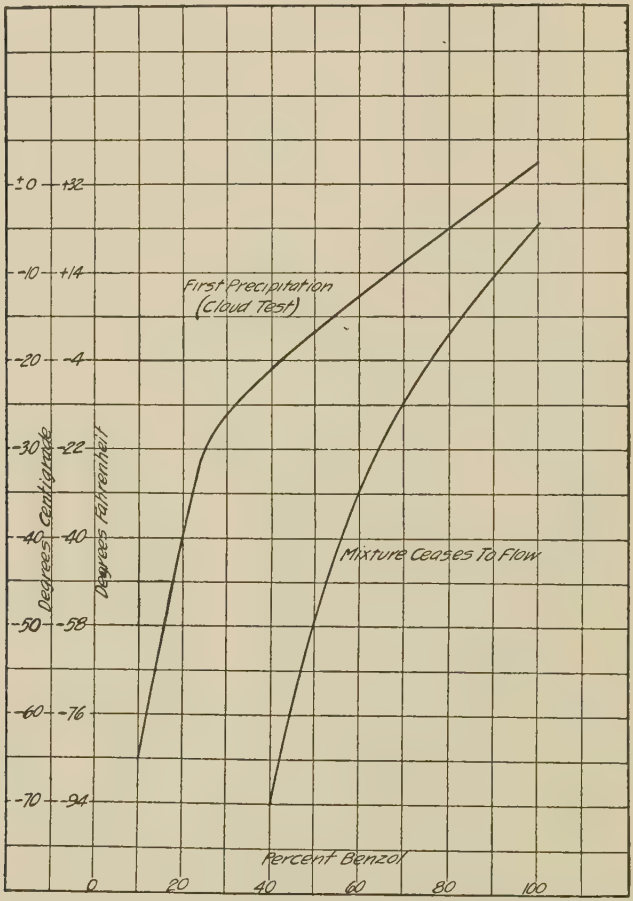


Figure 37. Freezing Points of Benzol Aviation Gasoline Mixtures

out by moisture. Figure 36 gives the results of tests by the Bureau of Standards to determine the proportions of benzol and gasoline required for various compression ratios. Figure 37 gives the results of tests conducted by the Engineering

Division to determine the freezing point of gasoline-benzol mixtures. Care should be exercised in the use of benzol as commercial benzol is usually far from pure and is apt to contain compounds which attack metals used in the fuel system in the engine. Impurities present also cause considerable variations in the anti-detonating properties and the freezing point.

Hydrogen. The extremely high heating value of hydrogen, 60,000 B.t.u. per pound, has led to consideration of this substance as a fuel for aircraft engines. Since hydrogen tends to detonate violently, it is probable that some modification in the engine construction will be required to permit the use of pure hydrogen; such modifications probably could be worked out successfully. This main obstacle in connection with the use of hydrogen as a fuel lies in the difficulty of carrying a gaseous fuel and the fact that liquefaction temperature of hydrogen is extremely low. It should also be noted that while hydrogen can be liquefied, the density of the liquid is very much lower than that of gasoline. The use of hydrogen as a fuel on an airship has frequently been resorted to in order to avoid valving the gas thus reducing the lift as the liquid fuel is consumed. This has been accomplished by using hydrogen in combination with gasoline and has been quite successful. However, the introduction of helium and water recovery apparatus has rendered this method of operation unnecessary in America.

In connection with the use of hydrogen as a fuel, the popular impression should be noted that the power output of the engine might be brought up in proportion to the heating value of the fuel; this is incorrect since the power output of the engine depends not upon the heating value of the fuel, but upon the heat developed per cylinder full of charge. The relative proportions of hydrogen and air are such that the heat developed per full cylinder is only slightly greater than that

developed by the same volume of gasoline and air; hence, while the weight of fuel consumed could be materially reduced, the output of the engine would not be increased in like proportion.

Lubrication

While a great amount of mathematical and experimental work of the highest character has been done on the problem of lubrication, results have not yet been obtained which permit of a fully rational analysis of the bearing design in an aircraft engine. The best that can be done at present is to use the theoretical and experimental results obtained as a guide to the interpretation of purely empirical data. Neither the physical properties of the lubricants nor the laws governing the action of lubricants between the bearing surfaces are clearly understood. In general, the action of the lubricant appears to consist in preventing the interlocking of high points on the rubbing surfaces. Two distinct types of lubrication are recognized—that in which the separation of rubbing surfaces is complete, and that in which incomplete separation exists. Complete separation can be secured only with certain forms of bearing surfaces when operating at relatively high rubbing velocities. The best illustration of this type of lubrication is the ordinary high-speed journal. The rotation of the shaft tends to draw the oil into the region of the maximum bearing pressure and, if the speed is sufficient, a pressure will be established high enough to separate the rubbing surfaces completely. The condition of stability of this type of lubrication is dependent mainly upon the unit loads, the rubbing velocity, the viscosity of the oil and the dimensions of the bearings.

Incomplete lubrication usually exists where the form of the bearing is such that oil is not drawn in between the surfaces with sufficient rapidity to form a separating film. A very thin film of oil is maintained on both surfaces which greatly reduces the friction but does not by any means completely

separate them as in the case just discussed. The behavior of an oil under these conditions seems to depend more upon surface tension than on viscosity, although both properties have an important bearing upon the allowable load and velocity. The study of this type of lubrication is much more difficult than when complete separation occurs since, separation being incomplete, the actual condition of the surface and the film between them is subject to almost infinite variations. The engineer is forced to depend entirely upon the best existing practice in estimating the loads and bearing areas which may be successfully operated.

Lubricants. For practical purposes lubricants may be divided into two general classes known as organic oils and mineral oils. Strictly speaking, both oils are of organic origin but the term "mineral oils" is usually applied to oils obtained from crude petroleum, while the term "organic oils" is usually applied to oils of animal or vegetable origin. Oils are usually classified on the basis of their viscosity, as this physical property chiefly influences its behavior as a lubricant. The viscosity of all lubricants varies rapidly with temperature as indicated in Figure 38. These curves show the variation of viscosity of two typical mineral oils and castor oil. It will be noted that the variation of viscosity with temperature exhibits the same general tendency in each case. Other properties of interest are the flash point that determines the temperature at which the oil ignites, the pour point that determines the temperature at which the oil becomes so viscous that it ceases to flow, freedom from impurities, chemical stability, and acidity. Reference should be made to existing War Department Specifications for lubricating oils which set forth the requirements as well as the testing methods employed.

Viscosity. As stated above, viscosity is the physical property mainly determining the lubricating value of an oil. In general, with partial separation of surfaces, the higher the

viscosity the greater the load which may be carried; while with complete separation the load which can be carried varies directly with the viscosity if other factors are held constant.

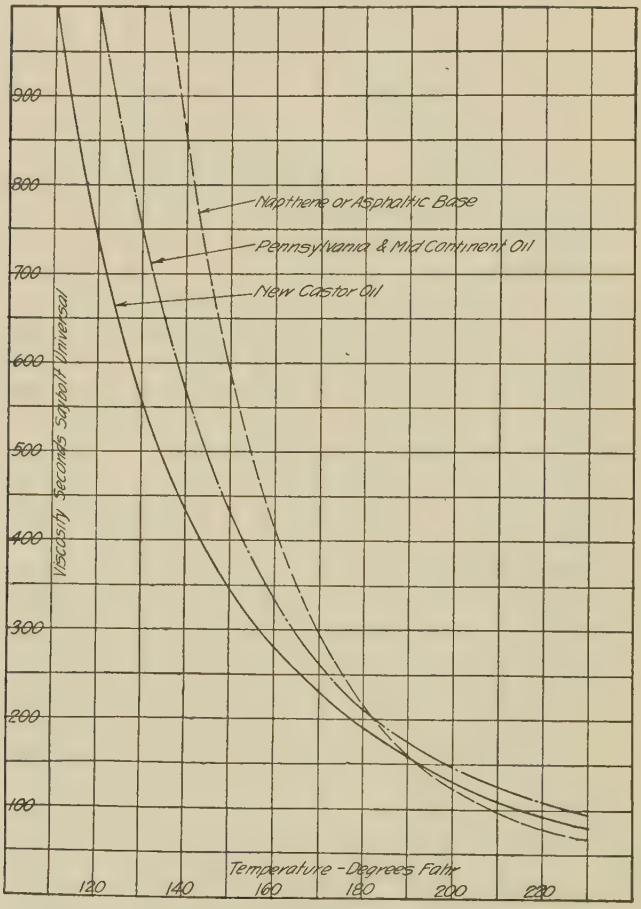


Figure 38. Variation of Oil Viscosity with Temperature

However, it should be noted that the heat developed also varies with the viscosity which tends to make viscous oil run hotter, thus reducing the viscosity itself. It should not, therefore, be concluded that bearing loads may be increased indefinitely

simply by increasing the viscosity of the lubricant. For very high speeds it is quite possible that a lubricant possessing lower viscosity at a given temperature will operate more satisfac-

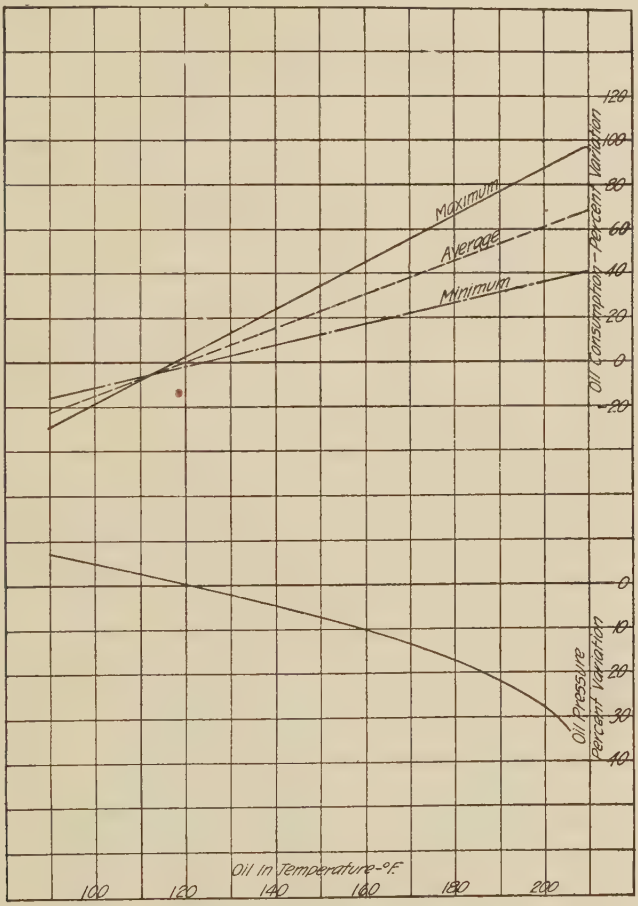


Figure 39. Effect of Oil Temperature on Oil Consumption

torily than an oil of higher viscosity, due to the smaller quantity of heat developed in the bearing. Hence, the cooling of highly loaded bearings has a very important influence on their ability to perform satisfactorily. The viscosity of the oil also has

an important effect on the total oil consumption of the engine. Figure 39 gives the results of some tests by the Engineering Division of the Air Service to determine the relation between oil consumption and oil temperature. The viscosity of the oil falling rapidly with temperature rise results in a rapid increase in oil consumption. Much the same results would be secured if tests were conducted with oils of various viscosities operating at the same temperature. This relation between the viscosity and oil consumption has led to a tendency to utilize extremely heavy oils for aircraft lubrication and this tendency appears to be sound within reasonable limits. However, extremely viscous oils are very difficult to handle in winter, as at low temperatures they do not flow readily through the oil passages, thus there is danger of lubrication failing completely. Starting under such conditions is very difficult. The higher viscosity oils are usually more expensive and the total supply available is somewhat restricted. For engines of average design a viscosity of 100 seconds Saybolt at 212°F. appears to be suitable for summer use, while a viscosity of 75 seconds Saybolt at 212°F. is to be preferred for winter use. For extremely cool weather, lighter oils having a viscosity of 50 seconds Saybolt at 212° F. have been successfully used; these grealty facilitate starting and warming up the engine.

Castor *vs.* mineral oils. As stated above, while viscosity undoubtedly has the greatest effect on the lubricating property of an oil, it has long been recognized that another property exists which considerably influences the value of the lubricant. This property is not well understood, but its existence is undoubted and it is usually referred to as "oiliness." This property is of more importance in the case of incomplete separation than when the surfaces are completely separated by an oil film. This superiority of certain oils in incomplete separation appears to be entirely independent of viscosity or other recognized physical properties. There appears to be no doubt that

organic oils possess this characteristic to a greater degree than mineral oils, and it is usually recognized that castor oil possesses lubricating values superior to those of the mineral oil of equal viscosity. Castor oil appears to possess a further advantage in that it suppresses corrosion more effectively than mineral oils. On the other hand, there are several disadvantages in connection with the use of castor oil, among which might be mentioned: scarcity and high cost, chemical instability, and the fact that it is not obtainable in a wide range of viscosities. With reference to chemical instability, the oil exhibits a tendency to deteriorate in storage and prolonged heating tends to develop gummy products which, if allowed to accumulate in sufficient quantities in an engine, are apt to be troublesome.

Many of the disadvantages in connection with the use of castor oil may be eliminated and most of the advantages retained by resorting to mixtures of castor and mineral oils. However, the possibilities in this direction are rendered less attractive by the fact that castor oil does not blend with mineral oil at ordinary temperatures, and if blended at higher temperatures will separate on cooling. In general, the European tendency appears to be toward the use of castor oil or blends of castor and mineral oil, while in the United States the tendency has been toward insisting on engines designed so that they may satisfactorily operate on straight mineral oil; it may be safely stated that the latter practice has not adversely affected the development and reliability of American engines.

CHAPTER VI

AUXILIARY SYSTEMS

Cooling Systems

Cooling systems represent a considerable portion of the power plant weight. Under this head are included the radiator water piping and connections, reserve water tank, shutter and shutter control, together with all water in the system.

Radiator types. Aircraft radiators may be classified under three types—the plate radiator, skin radiator, and the core radiator. The core type may be further subdivided into the ribbon type and the cellular type.

The plate radiator consists of a number of cooling elements in the form of hollow plates carrying water in a thin sheet. The plates are connected to suitable headers and a sufficient number are provided to give the cooling surface required.

The skin type radiator consists of a cooling surface designed to be applied to the surface of some part of the body, thus utilizing for cooling purposes a surface which is necessary for other purposes.

Core type radiators employ a large number of small cooling elements which are so joined together as to form a solid core. In the ribbon type the water passages of each element are separated from each other so that water entering any passage has a definite path through the core. In the cellular type the water passages are inter-connected so that the path of the water through the core is not definitely determined.

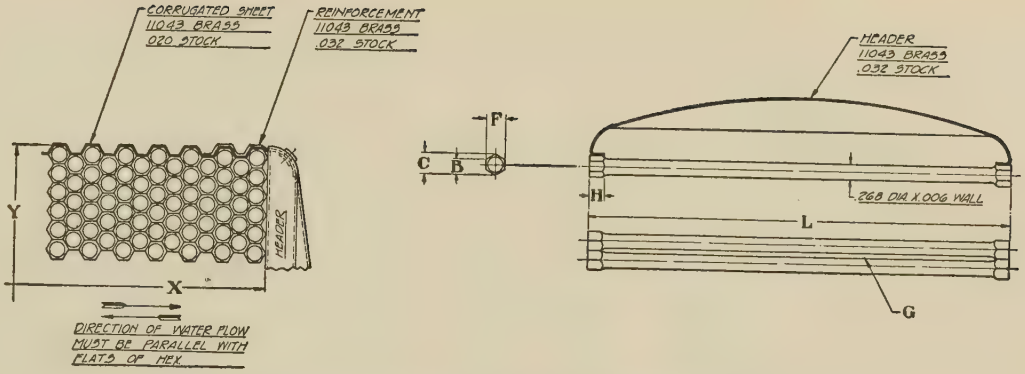
Comparison of types. As a basis on which the various types of radiators may be compared, the following desirable qualities are set forth:

1. Sturdy construction combined with low weight.
2. Low head resistance with high heat dissipation.
3. Small water passages without undue restriction to water flow.
4. Ease of maintenance and repair.

The core type radiators have the distinct advantage as regards strength of construction, light weight, and ease of repair and maintenance. The plate and skin type radiators are generally superior as to head resistance for a given heat dissipating capacity. On the other hand, if the weight be kept within reason they are of essentially flimsy construction and are usually difficult to repair, due to the inaccessibility of many of the joints and seams. Efficient shuttering of these types is also very difficult.

Honeycomb radiator. A particular type of cellular core radiator, commonly known as the honeycomb or cartridge type radiator, possesses such distinct advantages from all points of view, that its use has become practically universal in American practice.

The core of this radiator is made up of a large number of thin walled tubes whose ends are expanded into hexagonal shape for a short distance. The tubes are stacked so that the hexagonal ends nest compactly and these ends are then soldered together. This type of core construction is illustrated in Figure 40. This type of construction has many excellent points in its favor. The tubular form of the individual elements permits great strength with very thin material. The water passages are comparatively small, thus keeping the weight of water contained in the radiator within reasonable values. The passages are inter-connected in such a way that an obstruction at any one point does not put a considerable portion of the water passage out of action, as the water can flow around the obstruction through the adjoining passages. All soldered joints in the core are on the outside and are therefore



L	F	B	C	H	A	N	S'	S	V	W'	W	E	G
LENGTH OF TUBE	ACROSS FLATS	$\frac{\sqrt{3}}{2}$ ACROSS CORNERS	ACROSS CORNERS	LENGTH OF HEX.	FRONTAL AREA OF TUBE	Nº TUBES PER SQ FT. FRONTAL AREA	SQ IN. COOLING SURFACE OF TUBE	PER SQUARE FOOT FRONTAL AREA: SQ FT. COOLING SURFACE	CU IN WATER SPACE	WEIGHT OF WATER	WEIGHT OF TUBES	SQ FT. AIR AREA	SQ IN. ACTIVE CORE TUBES
4	.313	.2711	.3614	.187	.08484	1697	330	38.95	178	6.3	15.9	.606	102
5	.313	.2711	.3614	.187	.08484	1697	411	48.43	223	8.0	19.8	.606	208
6	.308	.2667	.3556	.250	.08215	1753	493	60.07	247	8.9	23.6	.626	220
7	.308	.2667	.3556	.250	.08215	1753	574	69.86	292	10.5	27.6	.626	260
9	.308	.2667	.3556	.250	.08215	1753	735	89.49	382	13.8	35.6	.626	340

$A = FB$

$N = \frac{144}{A}$

$S = [3(C-.0138)2H]+804(L-2H)$

$S = \frac{NS'}{144}$

$V = (144 - .0364N)(L-2H)$

$W' = .036V$

$W = $ BUREAU OF STDS. (INCLUDES SOLDER)

$E = \frac{.0314N}{144}$

$G = (F-.268)(L-2H)$

$Y =$ DIMENSION OF CORE PERPENDICULAR TO WATER FLOW

$N_Y = \frac{Y}{F} =$ NUMBER OF TUBES PER ROW

$X =$ DIMENSION OF CORE PARALLEL TO WATER FLOW

$N_X = \frac{X-C}{B} + 1 =$ NUMBER OF ROWS IN DISTANCE X (ODD NUMBER REQ'D WITH ABOVE DESIGN)

$N_T = N_Y N_X - \frac{N_X - 1}{2} =$ TOTAL NUMBER OF TUBES IN CORE

Figure 40. Details of Honeycomb Type Radiator

readily accessible for repair, and in case a failure occurs in the wall of a tube, the single tube may be easily removed, without unsoldering other portions of the core. After repair this type of core is as good as new, a feature which is almost invariably lacking in other types. The construction is such that any desired variation of frontal area and tube length may be employed. The ratio of heat dissipation to frontal area of this type of radiator, while not as good as some types of plate radiator, is still highly satisfactory.

Radiator headers and side walls. The method of making up the side walls for the honeycomb core is illustrated in

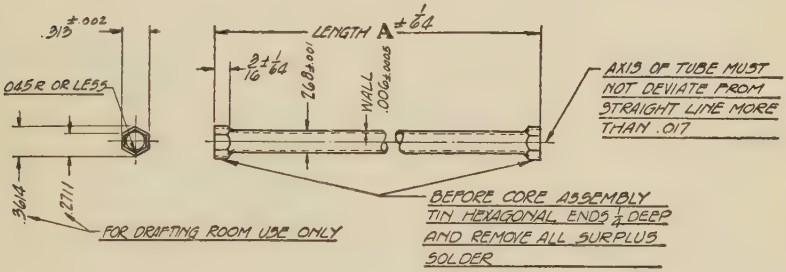

.313 ±.002
.045R OR LESS
.3614
.2711
FOR DRAFTING ROOM USE ONLY
LENGTH A ±1/64
2 1/16 ±1/64
.268±.001
WALL .006±.003
AXIS OF TUBE MUST NOT DEVIATE FROM STRAIGHT LINE MORE THAN .017
BEFORE CORE ASSEMBLY TIN HEXAGONAL ENDS 1/4 DEEP AND REMOVE ALL SURPLUS SOLDER
FOR TUBES 5 IN LENGTH OR LESS

Figure 41. Details of Short Honeycomb Radiator Tubes

Figure 40. This arrangement has the advantage of giving a very stiff construction while employing rather light metal. It has the further advantage of fitting snugly to the radiator core without forming a by-pass for the water along the side of the core. The inlet and outlet headers should be designed with ample capacity to permit uniform distribution of water entering and leaving the core. They should also be so designed as to withstand considerable pressure and ordinary strains due to mounting, handling, etc. These requirements may best be met by avoiding flat surfaces and using rounded shapes wherever possible. In general, care should be taken to avoid attaching to the radiator or headers projections in the form

of lugs, studs or tubes which are easily damaged in handling the radiator prior to installation. Figure 40 also gives complete information as to the weights, water capacity and cooling surface of radiators built up from honeycomb core. Figures 41 and 42 give the detailed dimensions of the individual tubes.

Shutters. In order to control the temperature of the cooling water, some form of shutter to prevent air flow through the radiator core is necessary. These shutters should be so designed as to cut off the air flow completely in order that the

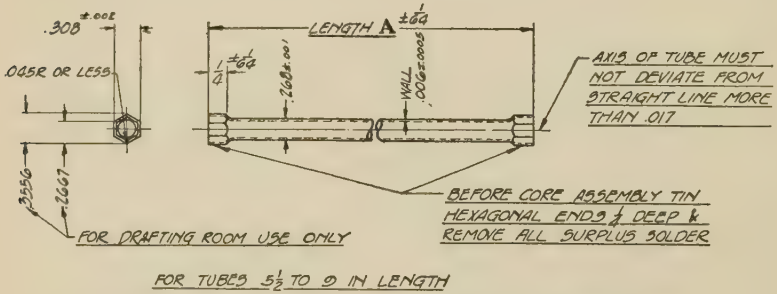
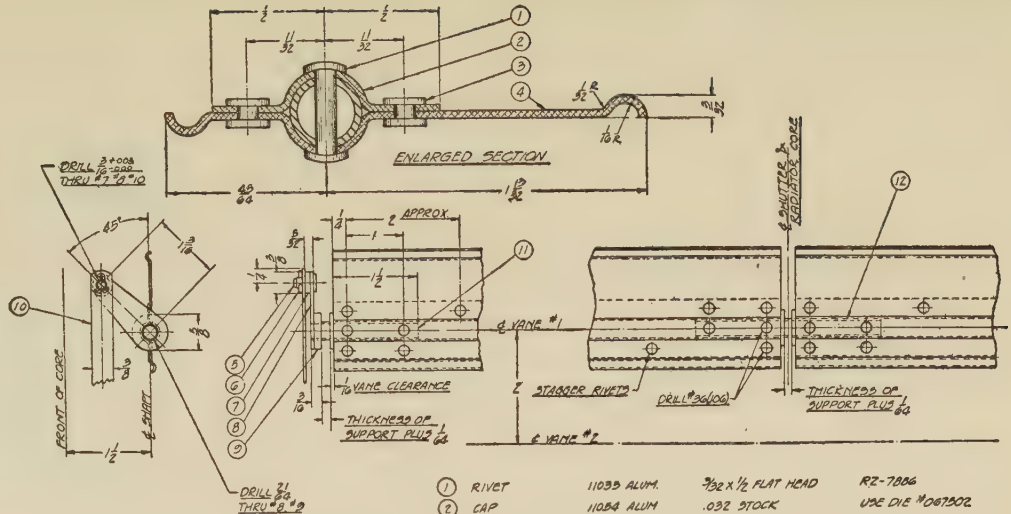

.308 ±.002
.035R OR LESS
.3556
.2667
FOR DRAFTING ROOM USE ONLY
LENGTH A ±1/64
±1/64
.268±.001
WALL .006±.0005
AXIS OF TUBE MUST NOT DEVIATE FROM STRAIGHT LINE MORE THAN .017
BEFORE CORE ASSEMBLY TIN HEXAGONAL ENDS 1/4 DEEP & REMOVE ALL SURPLUS SOLDER
FOR TUBES 5 1/2 TO 9 IN LENGTH

Figure 42. Details of Long Honeycomb Radiator Tubes

engine may be warmed up quickly in cold weather, and to permit maintaining the water at a suitable temperature when operating at low power outputs. The window blind type of shutter is generally employed with the core type radiator. Figure 43 shows some details of a shutter of this type which has proved very satisfactory.

Reserve water. A small amount of reserve water actually in the circulating system is desirable in order to make up for evaporation losses which cannot be completely avoided. The system should be so designed that this reserve water is above the highest point in the engine that requires cooling, under all normal conditions of operation. Besides the reserve water actually in the circulating system, an additional reserve may



(1)	RIVET	11033 ALUM.	3/32 x 1/2 FLAT HEAD	RZ-7806
(2)	CAP	11084 ALUM	.032 STOCK	USE DIE #067502
(3)	RIVET	11033 ALUM.	3/32 x 3/16 FLAT HEAD	RZ-4438
(4)	VANE	11084 ALUM	.032 STOCK	USE DIE #067524
(5)	CLEVIS PIN	10045 STEEL	3/16 x 1/4	PC-1519
(6)	COTTER PIN	STEEL	1/16 x 3/8	PR-328
(7)	SPACER	10201 STEEL	.0938 STOCK	WD-1849
(8)	ARM	10201 STEEL	.0625 STOCK	
(9)	HUB	10201 STEEL	.125 STOCK	WD-1826
(10)	LINK	10201 STEEL	.050 STOCK	
(11)	OUTER SHAFT	10225 STEEL	5/16 O.D X.035 WALL	
(12)	INNER SHAFT	10225 STEEL	5/16 O.D X.035 WALL	

TACKWELD & DIP BRAZE SPEC #10003
ZINC PLATE SPEC #8111 AFTER BRAZING
DURAL OR EQUIVALENT—
ANNEAL BEFORE FORMING AND TEMPER AFTER UPSET ALL DURAL RIVETS WITHIN 45 MINUTES AFTER TEMPERING
FINISH 2 COATS VARNISH SPEC #10001

Figure 43. Details of Window Blind Type of Radiator Shutter

be carried in a separate tank, in which case means for introducing this water into the cooling system is required. In this connection, provision for shut-off valves in the pipes between the radiator and engine is desirable in order that the engine may be drained, if necessary, without draining the entire cooling system.

Heat dissipation. The quantity of heat dissipated to the cooling water by various engines is remarkably uniform, usually lying between 22 and 26 B.t.u. per delivered horsepower per minute. In case the exact figure is not available from test, a value of 25 may safely be assumed.

Cooling capacity. The cooling capacity of a given radiator is dependent upon the temperature differences available, the rate of water flow and the rate of air flow over the cooling surface. For practical purposes the quantity of heat dissipated varies directly as the temperature difference and directly as the rate of air flow. The heat dissipation is not materially affected by the rate of water flow, provided this rate is sufficiently rapid to induce truly turbulent flow. Figure 44 gives the quantity of heat dissipated by the honeycomb radiator at various air speeds for the 5-, 7- and 9-inch tubes. It should be carefully noted that these curves are based on the assumption of unobstructed air flow. Obstructions either before or behind the core rapidly reduce the rate of air flow through the core; thus seriously cutting down the rate of heat dissipation.

Cooling system weights. Inasmuch as heat dissipation of a radiator varies rapidly with the velocity of air flow, the total cooling system weight varies considerably between installations on various types of aircraft. It being impossible to operate an airplane at flying speed below a definite minimum, it is safe to assume a considerable velocity of the air approaching the radiator under all flight conditions. On the other hand, certain maneuvers of airships require practically full engine power at or near zero air speed, hence greater cooling capacity

for a given horsepower is usually provided on this type than is the case of airplanes. The cooling system weight for airplanes, including radiator, shutter and water in the radiator,

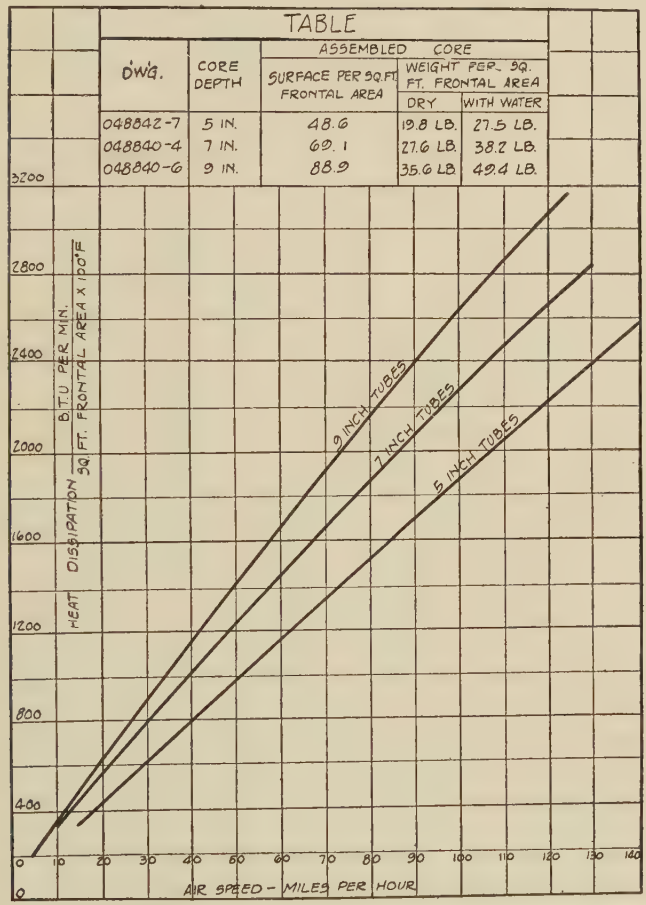


Figure 44. Heat Dissipation of Honeycomb Type Radiator at Various Air Speeds

usually lies between .55 and .8 pounds per horsepower, while for airships the weight is seldom below .7 pounds per horsepower and may be as high as 1.2 pounds per horsepower. Fig-

ure 44 gives the weight characteristics of the 5-, 7- and 9-inch honeycomb radiator.

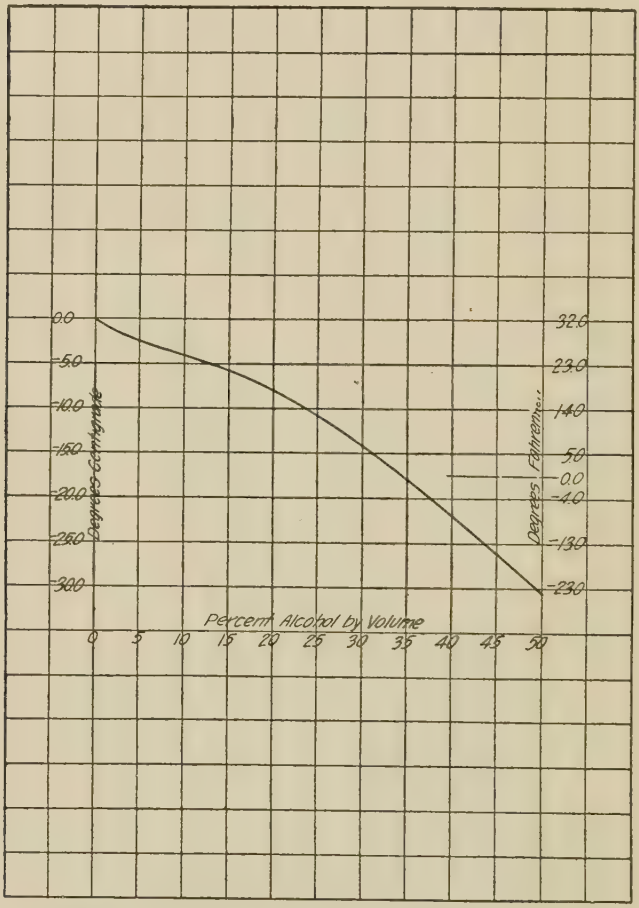


Figure 45. Freezing Points of Alcohol-Water Mixtures

Anti-freeze solutions. Operation in cold weather frequently requires the use of some medium in the cooling system which does not freeze at temperatures likely to be encountered. Alcohol is most commonly used for this purpose. Figure 45

gives the freezing point of various proportions of alcohol and water, while the specific gravity of these mixtures is indicated on Figure 46. Alcohol has the advantage of being readily

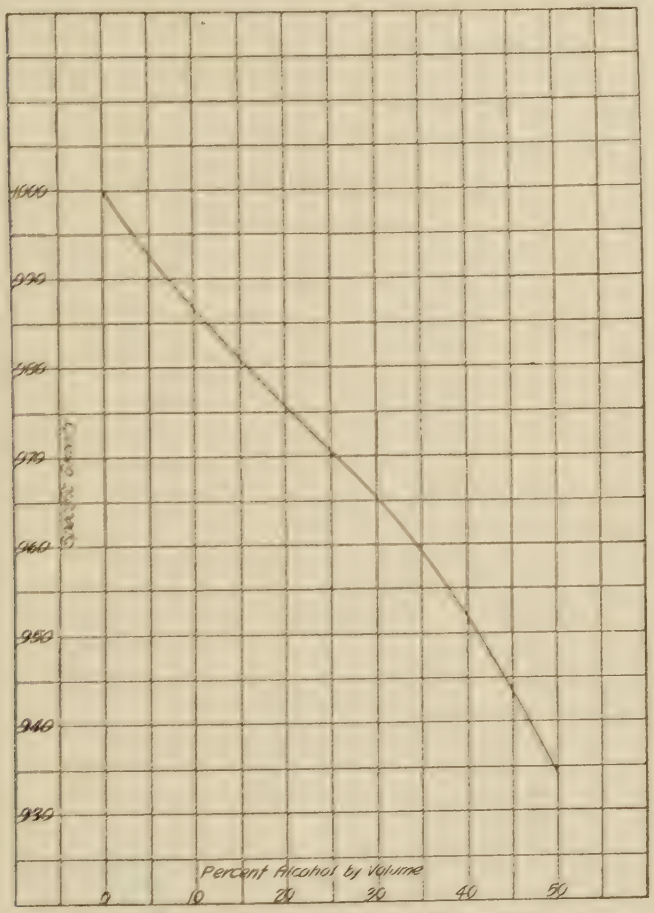


Figure 46. Specific Gravity of Alcohol-Water Mixtures

available, mixing easily with water and having no tendency to attack chemically any portion of the cooling system. On the other hand, it possesses the disadvantage of lowering the boiling point; however, this is usually not serious as the cool-

ing medium seldom reaches maximum temperatures during cold weather operation. There are numerous other substances suitable for cold weather cooling, such as glycerine and certain salts. Space does not permit a discussion of the properties of the various substances which might be used, but care should be taken in the selection of a cooling medium to insure that solids are not deposited in the cooling system, and that the metals in the system are not attacked by the substance used.

Fuel Systems

The aircraft fuel system varies according to the size and types of installation from a very simple arrangement consisting of a single tank with feed pipe and shut-off valve to a highly complicated system employing a multiplicity of tanks, elaborate feed lines, transfer pumps and numerous control valves. In large airships the location of the fuel load has an important bearing on the structural design of the ship and considerations of this nature usually require that this load be distributed between a number of tanks located at points advantageously selected throughout the structure. In the airplane the fuel load is usually located as near as possible to the center of gravity; this is desirable from considerations of structural strength, maintenance of equilibrium, and ease of control. However, as the development of very large airplanes progresses, it is quite probable that a system similar to that utilized in large airships may be required.

Tank construction. Terne plate, dairy tin, brass and aluminum are used in tank construction. Aluminum is most desirable from the weight viewpoint, but possesses the disadvantage that it is not easily soldered consequently small repairs are difficult to make. For this reason, the use of aluminum tanks is not recommended on small craft where the entire fuel load is carried in a single tank. Under these circumstances a slight

sacrifice in weight is justifiable in order to render it possible easily and quickly to repair any damage which may occur. On airships using a large number of separate tanks, inability to use a single tank is usually not of serious consequence and the total weight of tanks being high, the use of aluminum is desirable as a means of minimizing weight. In the design of gasoline tanks, irregular shapes employing large flat surfaces or re-entrant angles should be avoided, as these necessitate much heavier construction and are much more easily damaged. Cylindrical tanks with domed ends are inherently rigid and have a large ratio of capacity to surface area. The size of individual tanks should be kept within a reasonable limit, as extremely large tanks are difficult to construct and install or remove and replace in case repairs are necessary. A further disadvantage of large tanks lies in the fact that should a leak occur, a considerable portion of the fuel supply is apt to be lost. Two hundred gallons in a single cylindrical tank represent about the maximum convenient size. Each tank should be equipped with a water-trap for catching water that may get into the system accidentally. This is easily accomplished by simply projecting the outlet pipe a short distance above the bottom of the tank. Provision for draining water from the bottom should also be made.

Fuel lines. Copper, brass or aluminum tubes are suitable for use in fuel lines. Aluminum has proved quite satisfactory and a considerable weight saving may be effected by its use. As in the case with tanks, it suffers the disadvantage of not being readily soldered, for which reason certain types of fittings can not be used. The allowable velocity in the fuel pipes depends to a certain extent upon the distance which the fuel travels through the pipe. In general, generous cross-sections should be provided in order to avoid the necessity of considerable pressures in the fuel line. The following sizes are recommended for short pipes, that is, up to 25 feet in length:

30 gallons per hour, 3/8 inch O.D.
60 gallons per hour, 1/2 inch O.D.
100 gallons per hour, 5/8 inch O.D.
150 gallons per hour, 3/4 inch O.D.

For longer pipes the cross-sectional area should be increased according to the length. The wall thickness for tubes carrying gasoline is largely determined by the necessity of being able to withstand ordinary handling or accidental blows from tools, etc., without injury. For tubes under 1/2 inch in diameter a wall thickness of 1/32 inch is recommended. For all larger sizes, 3/64 inch wall is desirable.

Connections and fittings. There are three different types of tube connections in common use; the flexible connection employing a short length of rubber hose, connections which are soldered to the tubes, and connections of the compression type which clamp to the tube without the use of solder. The rubber connection is usually employed only at points where considerable flexibility is necessary. This type of connection requires frequent inspection and periodic replacement. However, with proper attention, it is a very satisfactory type of connection as it is capable of withstanding considerable distortion and its deterioration is always gradual, a sudden and complete failure being very unlikely. It is also very easy to change when replacement becomes necessary. The use of rubber connections in inaccessible places where inspection is difficult should be carefully avoided as they are sure to be overlooked and failure is bound to occur sooner or later. The use of beads on the tube end is not recommended for rubber connections on fuel lines due to the fact that the rubber hose is more likely to be injured in forcing it over the bead, and, since the bead stretches the hose, the hose fails much more readily when deterioration from the action of gasoline sets in. Figure 47 illustrates a connection of this type made up in accordance with standards used in the Army Air Service. The metallic liner is inserted for the purpose of preventing

particles of rubber entering the fuel lines; while this type of liner offers a slight restriction of flow, it is considered superior to external types which necessitate considerable stretching of the rubber hose.

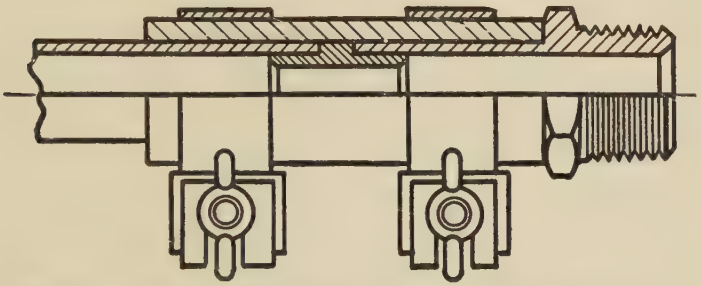


Figure 47. Section of Army Air Service Rubber Fuel Connection

Great care should be exercised in the selection of soldered or compression connections. Only types which have proved satisfactory in service should be employed without careful preliminary tests. The soldered type is generally preferred to the compression type because it lends itself better to frequent

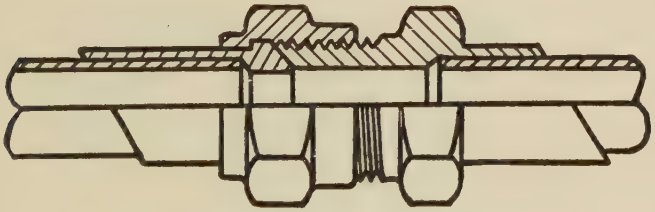


Figure 48. Section of Soldered Fuel Connection

dismounting. The portion soldered to the end of the tube forms a protection which prevents damage to the ends while being taken apart or reassembled; the section of metal at the point where the seal is made is fairly heavy and not easily damaged. This type of connection is not readily applied to aluminum tubes as soldering cannot be employed. However, by making of aluminum the portions which actually attach

to the tube, welding may be resorted to as a means of attachment. A common type of soldered connection is illustrated in Figure 48.

The compression type of connection offers the advantage that it may be employed with an aluminum tube without the necessity of welding. The type illustrated in Figure 49 has given excellent results in service. However, this type is not considered equal to the soldered type when frequent dismounting is necessary. The exposed ends of the tube are apt to be slightly bent which renders a tight joint difficult; then if ex-

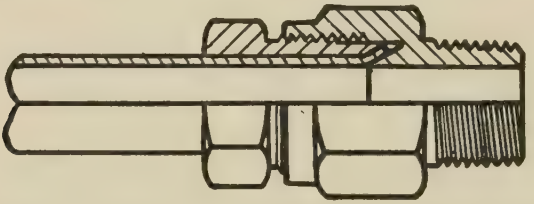


Figure 49. Section of Compression Type Fuel Connection

treme pressure is used in drawing up the connection, the tube is apt to be damaged with the stresses highly localized which may result in failure under slight vibration.

Fuel pumps. It is usually impossible to lay out an installation which permits the fuel to be delivered to the engine entirely by means of gravity, hence some type of fuel pump is necessary. Fuel pumps are employed for transferring fuel from one tank to another to alter the balance of an airship; also servicing pumps for filling the tanks on both airplanes and airships are frequently provided. Pumps supplying fuel from the tanks to the engines are usually engine-driven or wind-driven, while servicing pumps and transfer pumps are usually manually operated. Small electric motors for driving the fuel pumps have much to recommend them as their operation is independent of the engine and forward motion of the aircraft. On the other hand, the motors with the wiring,

switches, etc., in close proximity to the gasoline pump, constitute a fire risk which has prevented their general use.

Since the presence of gasoline renders the effective lubrication of the working parts of the fuel pump practically impossible, the design and construction of a satisfactory fuel pump requires the greatest care. Two types of pumps have been in general use; pumps employing plungers or oscillating vanes, and gear pumps. Pumps of the former type are usually operated at relatively low speed which permits fairly satisfactory results with unlubricated surfaces. The chief disadvantages of this type lie in the pulsating character of the discharge and the fact that the operation is dependent upon the proper functioning of a number of check valves. Being essentially suited to low speed, this type of pump is usually employed for hand operated pumps. Gear pumps possess the advantage that no valves are required and since they may be operated at fairly high speeds, they are less bulky and lighter than the plunger type. High-speed operation demands that extreme care be exercised in construction to secure proper clearance and fits throughout. However, when properly designed and carefully constructed, pumps of this type have given excellent results.

Strainers. Minute particles of solid matter in the fuel are apt to interfere seriously with the operation of fuel pumps and carburetors. For this reason, fine mesh strainers should be installed in such a manner as to protect these devices from damage. A water-trap is usually included in the strainer design, although it has been demonstrated that a fine mesh screen cannot be relied upon to separate the water from the fuel when moving at any considerable velocity. A screen of 60-mesh bronze or monel metal is recommended. A sufficient area should be provided to avoid a considerable drop in pressure through the screen. An area of 15 sq. in. for each 50 gallons per hour flow is considered sufficient.

Connecting systems. The general arrangement of the connecting system of fuel pipes for airships differs considerably from that usually employed in airplanes. This is due principally to two differences in the operating conditions. In the lighter-than-air type the various elements of the fuel system are usually accessible during flight and ordinarily are not under the direct control of the pilot operating the airship. In the heavier-than-air type the control must generally be in the hands of the pilot, and the various parts of the system are not usually accessible during flight. For these reasons in airships it is customary to duplicate important fuel lines and to provide numerous shut-off valves for isolating any line in which failure develops, whereas on airplanes duplication of fuel lines is seldom resorted to, as in most cases it would be impossible to isolate a defective line. Furthermore a multitude of cocks so arranged as to be under the immediate control of the pilot leads to great complication and much additional piping which more than offsets the advantages to be gained from duplication. As mentioned above, it is probable that as the airplane development progresses to larger sizes the general arrangement of the fuel system will tend to approach that employed on large airships.

In large airships the fuel is usually carried in a number of tanks distributed throughout the structure. A proper system of connection requires that any engine may receive fuel from any tank. Furthermore, considerations of balance necessitate some provision for transferring fuel from one tank to another. This is best accomplished by providing a common feed line with branch connections to each engine and to each tank. Each of these branch connections should be provided with a shut-off valve so that any desired combination of engine and tanks may be utilized. In order to transfer fuel for purpose of balance, one or more transfer pumps are connected to the main feed line in such a manner that the inlets and outlets may be reversed; that is, fuel may be pumped

along the feed line in either direction. A shut-off valve is required in the feed line at the point where the transfer pump is connected. The main feed line also may be used for filling the tanks, and by opening the proper valves any desired combination of tanks may be filled or emptied.

In the airplane it is desirable to restrict to a minimum the number of elements entering into the fuel system. The gasoline usually is located near the center of gravity, hence means for transferring fuel from one tank to another are not required. The control system must be as simple as possible and a multiplicity of tanks should be avoided. The fuel system of a multi-engine airplane offers a considerable problem in balancing the advantages of great flexibility with the additional safety thereby introduced, against the probabilities of trouble incurred through the introduction of greater complication involving elements which constitute possible points of failure. Many arrangements are practicable, such as a centralized supply with common pump for all engines, a centralized supply with individual pumps for each engine, separate and independent supply and pump for each engine, separate supply and pump for each engine combined with means of inter-connecting supply tanks and pumps in case of emergency. The arrangement employing a central supply with separate pumps for each engine would appear to offer the balance of advantages in most cases. Complication in the control system and piping will be at a minimum; each engine having its own pump, the probability of failure in the system supplying any engine should not be greater than the probability of failure of any other part of the engine. Efforts to provide greater flexibility, such as arranging to supply one engine with fuel from a pump mounted on another, introduces many more pipes, connections and controls, resulting in a loss rather than gain in overall reliability.

CHAPTER VII

REDUCTION GEARS AND REVERSING MECHANISMS

Certain maneuvers of an airship require the ability to reverse the direction of the thrust of the air screw. Three means are available to this end—reversible engines, reversible gearing and reversible pitch propellers. The method best suited to any particular case depends to some extent upon other features of the system such as the location of the power plant, type of engine used, and whether or not reduction gears are to be employed.

Reversible engines. In order that a gas engine may operate with the direction of rotation reversed, it is necessary that mechanism be provided which causes sequence of valve timing events to take place in the same order as when operating in the normal direction. This may be accomplished by reversing the direction of rotation of the camshaft relative to the crankshaft, or by employing a different set of cams for reverse operation. The Maybach engines employ camshafts equipped with two sets of cams. By shifting the shaft longitudinally either set of cams may be brought into operation. In addition to insuring correct valve timing during reverse operation, provision must be made for the proper operation of accessories, such as oil pump, water pump, ignition, etc. These must either be of a type that operates satisfactorily when rotating in either direction or must be provided with a drive which permits them to rotate in the same direction regardless of the direction of the crankshaft rotation. This method of reversing the propeller thrust is very attractive from the point of view of weight and simplicity of installation. On the

other hand, it is open to the objection that it requires stopping and restarting the engine each time the propeller is reversed. This requires absolute certainty of operation of the starting system, a requirement not easy to assure under all conditions.

Reversing gears. The use of reversing gears as a separate unit is objectionable because of excessive weight. On the other hand, if a reduction gear is to be employed the addition of a reversing gear to this mechanism does not greatly increase the weight; hence, the use of reversing gears appears to be desirable only when a gear reduction or transmission is to be used.

Reversible propellers. The reversible propeller is discussed in Chapter X. Comparing this device with reversing engines or reversing gears, it has much to recommend it. The engine may be reversed without stopping and the additional weight is not great. No clutch is required—and the propeller revolving continuously serves as a very satisfactory flywheel while idling. The success that has been attained in this mechanism in small sizes leads to the opinion that a satisfactory reversible propeller in large sizes is merely a matter of further detailed development; it would appear that the reversible propeller should be seriously considered as a means of reversing the thrust.

Reduction gears. The basic requirement of minimum weight necessitates rather high crankshaft speeds for aircraft engines. It frequently develops that for a given combination of power and air speed, the rate of propeller rotation required for maximum efficiency differs considerably from the normal speed of crankshaft rotation. In such cases gearing may advantageously be introduced between the engine and the propeller. The laws governing the selection of a suitable rate of propeller rotation for any given combination of power input and speed of advance are discussed in Chapters IX and X. Reduction gears may be built in separate units or may be built

as an integral part of the engine. The latter arrangement possesses the advantage from the point of view of weight and is the one usually employed for airplanes, while considerations of convenient arrangement of the power plant together with the necessity for a reverse gear, render the separate unit more desirable for airships.

Transmissions. The use of two or more engines connected to a single propeller through a transmission is frequently desirable. The transmission usually contains a gear reduction and the combination of gearing for two or more engines into a single unit usually results in a considerable saving of weight. By the use of transmissions of this type, the designer is enabled to group two or more engines into a single unit which is advantageous from the operating point of view, and usually results in a more compact installation. Clutches are provided ordinarily so that in case of failure of one of the engines, it may be disconnected, allowing the operation of the unit with the remaining engines.

Gear design. The ever-present requirement of minimum weight renders the design of a satisfactory transmission or reduction gear for aircraft one of the most difficult problems in the gear-making art. The manufacturers who have attained the greatest success in this field have particularly guarded the design data and methods of calculations which they employed; information from other sources is of little value when applied to problems of this character. For this reason no data pertaining to gear design can be given here which may not be found in standard works devoted to the design and construction of high-grade gearing. In lieu of this information, brief descriptions of some typical cases are given.

Shenandoah reduction and reverse gears. Figures 50, 51 and 52 show the general arrangement of the reduction and reverse gears used on the U. S. S. "Shenandoah." A clutch of the multiple plate type is inserted between the engines and

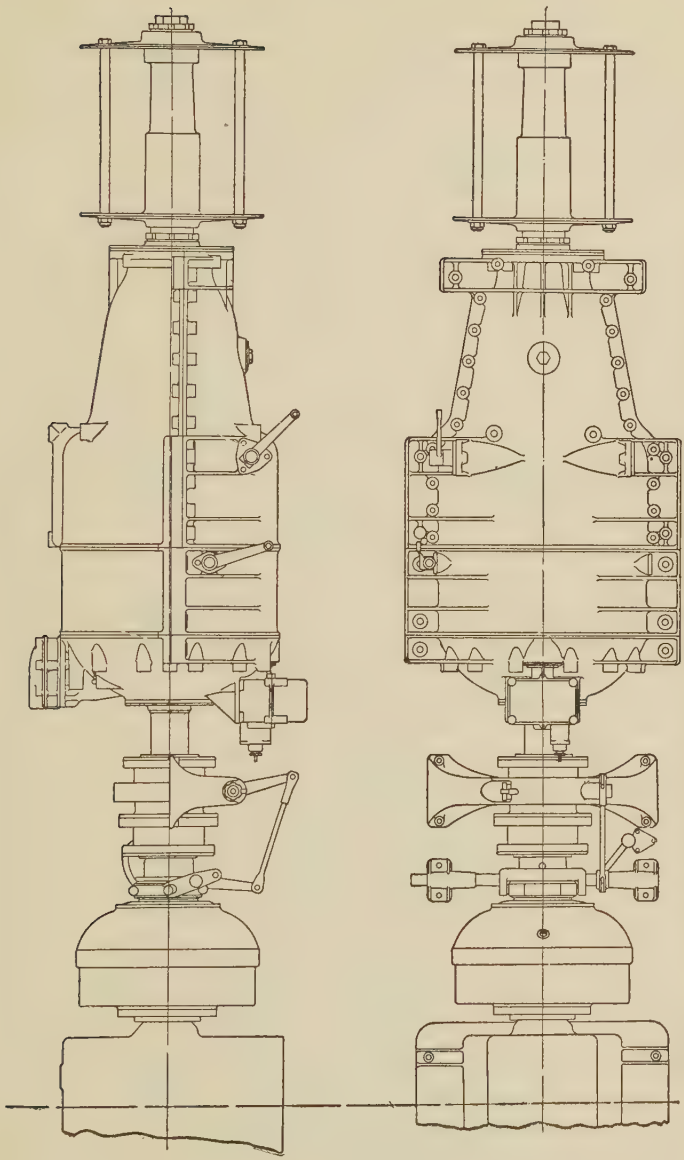


Figure 50. Plan and Elevation of U. S. S. "Shenandoah" Reduction and Reverse Gears and Clutch

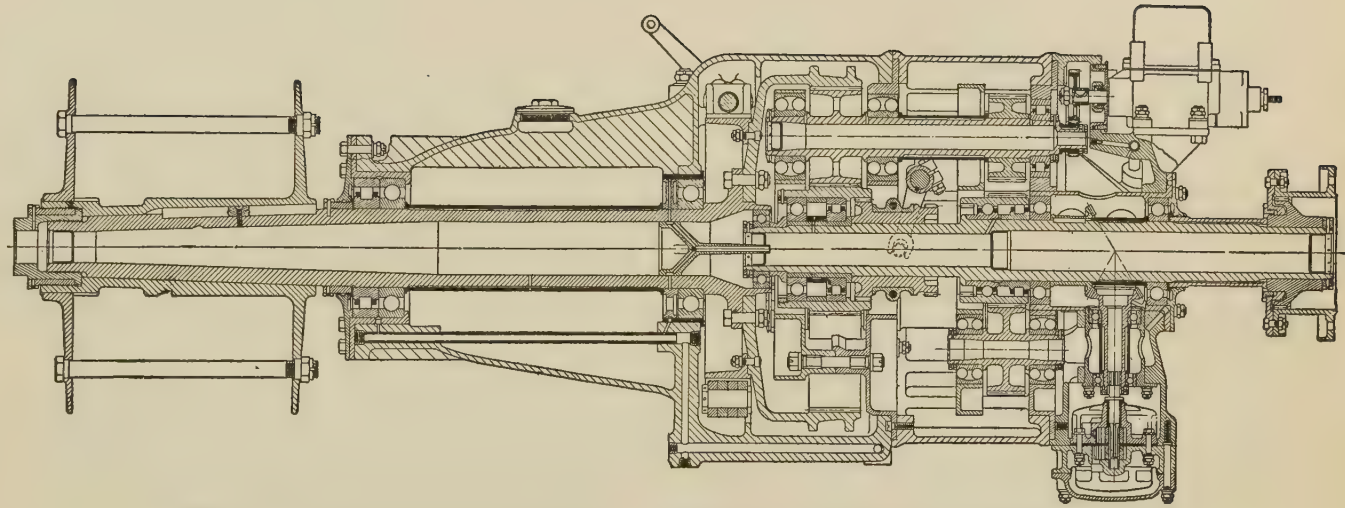


Figure 51. Longitudinal Section of U. S. S. "Shenandoah" Reduction and Reverse Gears

the gear box, thus permitting the propeller to be stopped while the engine continues to run. A brake is provided which enables the operator to stop the propeller in a horizontal position when necessary for ground clearance. The gearing is of the epicyclic type; both direct and reverse gears being in constant mesh. The drive is taken through a sliding dog which con-

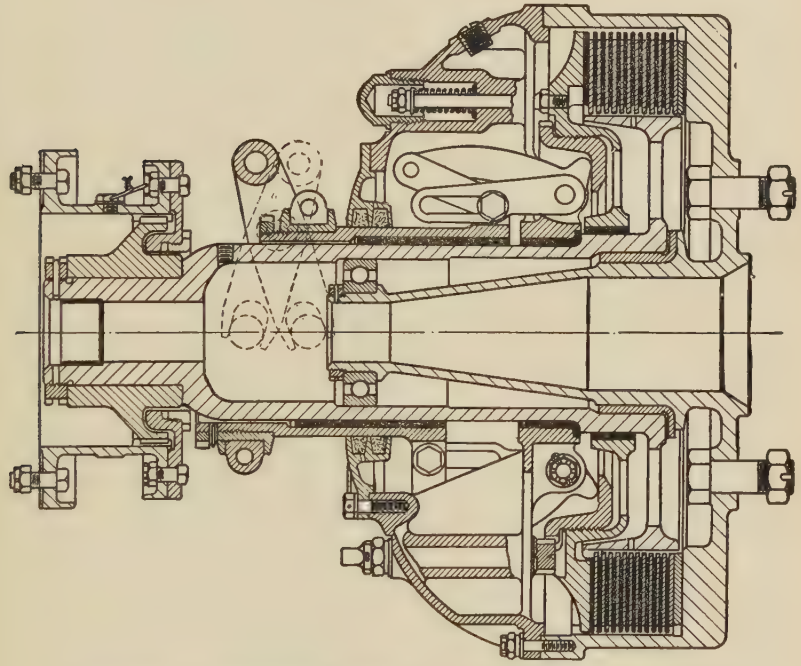


Figure 52. Longitudinal Section of U. S. S. "Shenandoah" Clutch

nects the drive shaft to either the forward or reverse gears, according to the position. This arrangement releases the working teeth of the gear from wear and tear incident to shifting, thus avoiding the danger of broken or burred corners in case the shift is improperly manipulated. Subsequent to installation it was found desirable to add a brake (not shown in the figure) to the driving member between the clutch and

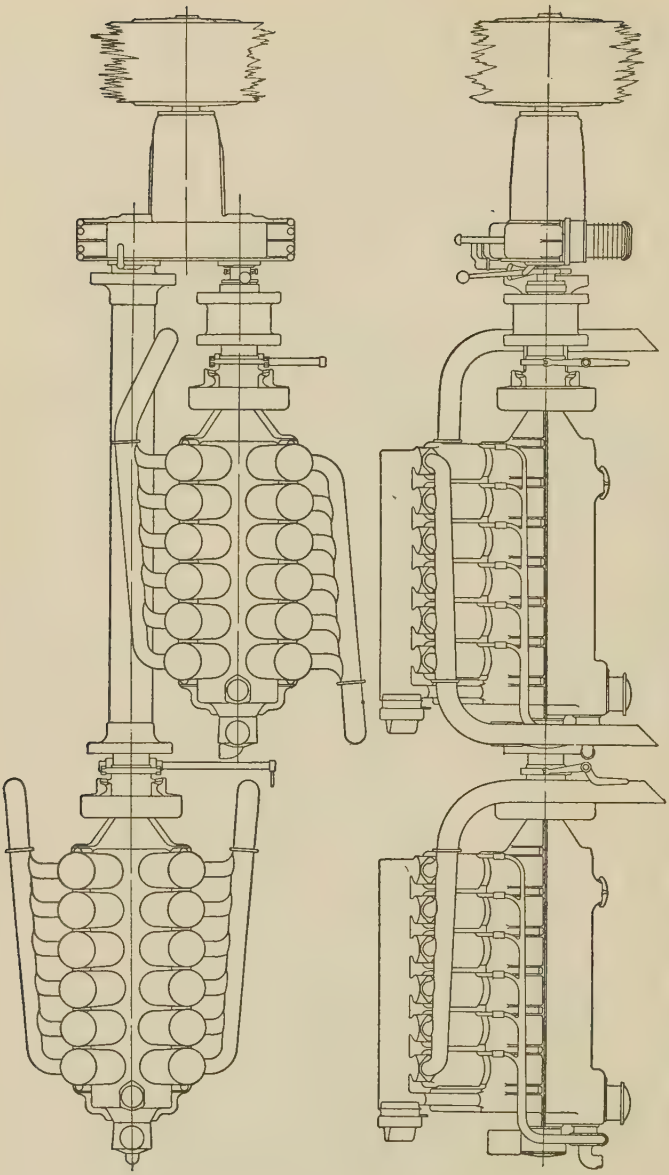


Figure 53. Plan and Elevation of Transmission Constructed for U. S. Army Type RS-1 Semi-Rigid Airship

the transmission for the purpose of stopping this member when it was desired to shift the gear. Without the brake this member tended to spin continuously, thus making engagement rather difficult.

RS-1 transmission. The transmission constructed for the Type RS-1 Semi-Rigid Airship of the United States Army, illustrates the type of power plant in which two engines are connected to a single propeller. The general arrangement of this transmission is indicated in Figure 53. The forward engine connects to the gear box through a disengaging clutch and a short length of drive shaft with a flexible coupling. The rear engine connects to the gear box through a clutch and coupling with a comparatively long drive shaft. Provision is made in the gear housing for reversing the rear engine only, as it is considered that one engine will provide sufficient thrust for operation in the reverse direction. The gear ratio is 12:5. The clutches are of the multiple-plate dry type and are engaged or disengaged by means of levers shown in the illustration. Either engine may be operated separately or both engines in unison and one engine being in operation may be used to start the other engine. The weight of the entire transmission, including clutches and drive shaft, is approximately 850 pounds.

CHAPTER VIII

SUPERCHARGERS

The use of superchargers on airplanes for military purposes is becoming quite common, while their use on airships and in commercial aircraft of both types, is a future possibility. Supercharger development is still in a relatively elemental phase and it is not yet clear what type or types will eventually be used. Several types are now under experimental development, but more experience will be necessary to determine the ultimate lines along which this development will progress. For this reason the following discussion is confined to general considerations.

Superchargers have been developed in order to reduce the drop in power resulting from engine operation at high altitude. The supercharger consists of a device for compressing the atmospheric air before it enters the engine cylinders, thus offsetting the drop in density due to altitude and permitting the same weight of charge to be delivered to the engine cylinders at high altitude as is taken in at sea level. Mechanically, a supercharger consists simply of an air pump with a suitable drive. Of the various types of air pumps available, the ones best adapted to supercharger service are those which are capable of handling a given volume of air with minimum weight and bulk. Two types of air pumps have been extensively used known as the Roots blower and the centrifugal air compressor. The former possesses the advantage of being able to deliver a large quantity of air when operating at moderate speeds; its size and weight are not excessive. The chief disadvantages are the pulsating character of its discharge and the difficulty of maintaining sufficiently small clearances in its construction to permit high efficiency when operating against considerable

pressure difference. The centrifugal compressor is rather more compact than the Roots blower; its discharge is free from pulsation; the efficiency is satisfactory throughout the desired pressure range, and mechanically it is very simple, the impeller being the only moving part. The chief disadvantage of the centrifugal compressor is that it must operate at a very high rotative speed, which tends to complicate the driving mechanism.

Superchargers have been driven by gearing them to the crankshaft of the engine and by providing a turbine operating from the engine exhaust. The Roots type compressor lends itself well to a gear drive, as the best speed of operation is not very different from that of the engine. The centrifugal compressor must be operated at speeds in the vicinity of 20,000 r.p.m. and thus requires a rather elaborate train of gearing when driven directly from the engine crankshaft. As yet no entirely satisfactory arrangement of this kind has been developed whereas little difficulty has been experienced in driving the Roots type compressor through a gear train.

The exhaust gas turbine operates best at a speed comparable with the speed of the centrifugal compressor and with this type of drive no gearing is required. The turbine and compressor rotors are mounted on the same shaft and bearings rendering the combination extremely simple from the mechanical point of view, the shaft with the two rotors being the only moving part. Superchargers of this type have been developed to a very satisfactory stage by the General Electric Company and a considerable number of them have been in use by the Army Air Service. The French have also developed a supercharger of this type and have placed a considerable number of them in the military service.

Comparing the turbine driven type of supercharger with the gear driven type, the former possesses the advantage of mechanical simplicity and ease of control. It may be adapted to any type of engine without special provision for attaching

a gear train and the supercharger mounting may be made entirely independent of the engine. The speed of the supercharger is not directly controlled by the speed of the engine and this type of supercharger may be made to operate considerably above its normal rating by simply increasing the speed. The chief disadvantage of this type is that it necessitates a manifold for collecting the exhaust gases and delivering them to the turbine. This manifold operating at a temperature close to a red heat is difficult to keep free from leaks and cannot be housed within the engine cowl. The exhaust turbine also operates at a very high temperature and must be placed outside the cowling in order that satisfactory cooling may be provided. The gear driven supercharger operating independently of the exhaust system may be entirely inclosed within the airplane structure and requires no modification of the normal exhaust system of the engine. The gear ratio being fixed, the amount of supercharging that is possible depends upon the rotative speed of the engine; there is no possibility of forcing the supercharger by overspeeding as can be done with the turbine type. Power to drive the geared supercharger is taken directly from the engine crankshaft and thus reduces the power available for turning the propeller, whereas the power used in driving a turbine supercharger is obtained from the expansion of the exhaust gases, thus utilizing energy which would otherwise be wasted.

Compression of the air in the supercharger results in a rise in temperature which becomes excessive in high altitude superchargers necessitating the introduction of an air cooler between the supercharger outlet and the air intake of the engine. Air cooler design follows essentially the same principles that are employed in radiator design. Like the radiator, the air cooler should be arranged to give the maximum heat dissipation for minimum weight and head resistance.

The increase in engine power secured by use of the supercharger necessitates larger water radiators and in order to pre-

vent boiling at high altitude a closed cooling system is required. This is usually accomplished by providing a relief valve in the cooling system which maintains a pressure of from 5 to 10 pounds, and permits steam to escape before a dangerous pressure is reached.

The fuel system of the supercharged airplane also presents special problems due to the fact that, at very high altitudes the fuel is apt to vaporize if the pressure is reduced further by suction from the fuel pump. This difficulty may be avoided by locating the fuel pump so that the fuel flows from the tank to the pump by gravity, thus avoiding any depression in the line between the pump and the tank. The fact that the fuel pump must deliver the fuel to the carburetor against the air pressure developed by the supercharger necessitates a special relief valve, as the head against which the pump delivers varies as the amount of supercharging is increased or decreased.

PART II

PROPELLERS

By

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CHAPTER IX

PRINCIPLES OF FLUID PROPULSION

The Aerodynamic Theory of the Propeller

The fundamental basis of fluid propulsion as derived from the laws of momentum. All actual means of obtaining propulsion in a fluid depend in the last analysis upon some method of setting in motion a mass of the fluid in a direction opposite to the motion of the craft to be propelled. In terms of the usual screw propeller as applied to aircraft it is necessary to blow backward a given mass of air in order to produce forward thrust or momentum to drive the craft through the air.

The thrust produced by a propeller is equal to the product of the mass of air handled during an interval of time by the velocity imparted to this mass of air. This law applies to the familiar slip stream which is a necessary and fundamental part of the propulsion system.

Figures 54 and 55 illustrate the simplest possible conception of the air flow through the propeller both in the case of the aircraft in motion and at rest.

The slip velocity may be computed from the following:—

$$T = \rho \frac{\pi D^2}{4}\left(V_1 + \frac{v}{2}\right) \times v$$

where T represents the thrust in pounds, D the diameter of the propeller in feet, V_1 the velocity of the aircraft, v the velocity of the slip stream. From this equation the slip-stream velocity may be computed when the propeller thrust and diameter and speed of the aircraft are known. The thrust may be computed from the following equation:

$$T = \frac{\text{hp.} \times 550 \times e}{V_1}$$

where e represents the propellor efficiency. It may also be shown that for a perfect propeller the limiting value of the efficiency is obtained from the following equation:

$$e = \frac{1}{1 + \frac{v}{2V_1}}$$

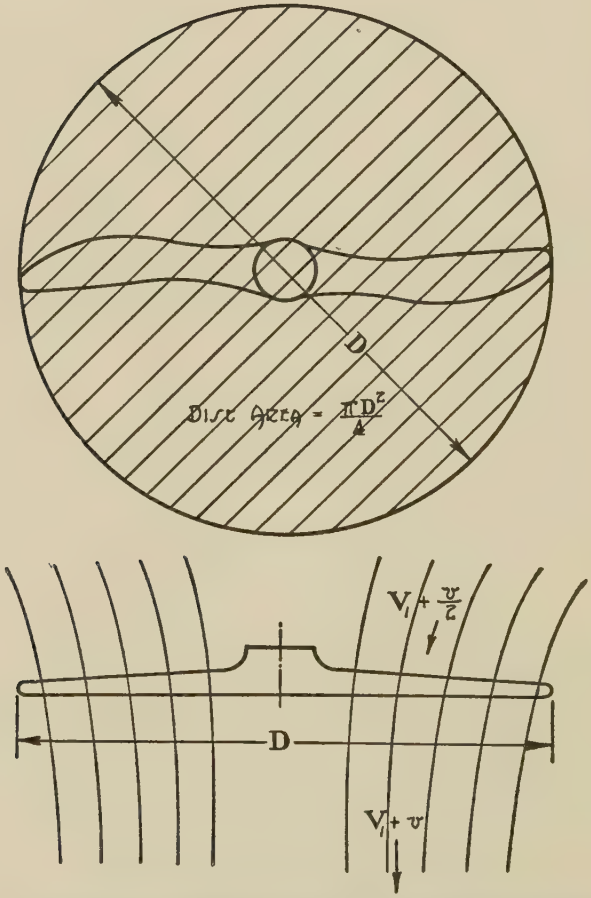


Figure 54. Elementary Picture of the Airflow Through the Propeller when the Aircraft is in Motion with a Forward Velocity of V_1

The above equations are obtained entirely from considerations of momentum and of energy in the air stream; the method of derivations may be found in any good textbook on air propellers.

The interesting point in connection with the equation for efficiency is to note that the limiting efficiency of the propeller depends only upon the ratio of the slip-stream velocity to the velocity of the aircraft. Obviously the lower the slip-stream velocity the higher will be the possible efficiency of the propeller.

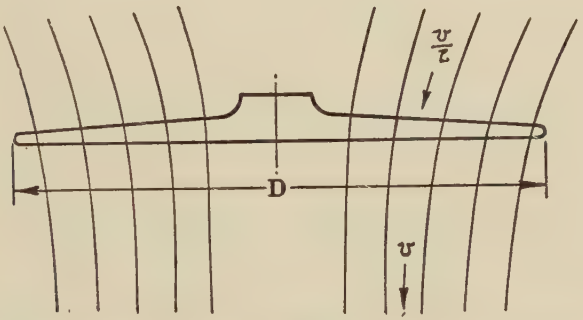


Figure 55. Elementary Picture of the Airflow Through the Propeller, When the Aircraft is at Rest

It will be seen from the equations connecting the thrust of the propeller with the slip-stream velocity that the slip-stream velocity depends on the diameter of the propeller; the greater the diameter, the lower the slip-stream velocity. Consequently from the standpoint of the theoretical limiting efficiency, the greater the diameter of an air propeller the greater will be its possible efficiency.

These equations are very fundamental in application and can be applied in judging the merits of any device for propulsion since the possibilities for efficiency depend entirely upon the mass of air to be handled by the propulsive device.

The same laws may be applied to the static thrust of the

propeller assuming that the velocity of the inflow in the slip stream is equal to one-half of the velocity in the outflow.

For conditions of static thrust, the following equations apply:

$$T = \frac{\rho . \pi D^2}{4} \cdot \frac{v^2}{2}$$

$$\text{Hp.} = \frac{\frac{\rho . \pi D^2}{4} \cdot \frac{v^3}{4}}{550}$$

$$\frac{T}{\text{Hp.}} = \frac{2 \times 550}{v}$$

From these equations it will be seen that the thrust per horsepower which can be obtained depends only on the slip-stream velocity. The slip-stream velocity in turn depends only on the pounds of thrust per square foot of disc area of the propeller.

A curve showing the relation between the pounds of thrust per square foot of disc area and the thrust per horsepower that can be obtained with an ideal propeller is shown by Figure 56. The thrust per horsepower which actually can be obtained from a propeller varies from 60 per cent to 90 per cent of the theoretical value according to the design of the propeller. The efficiency of the propeller as a thrust producer is usually considered as the ratio of the actual pounds of thrust per horsepower obtained to the theoretical limit as obtained from the curve.

Airfoil or zone theory of the propeller. In actual practice the theoretical efficiencies are never obtained, and considerable study has been given to the problem of predicting the efficiency, the power absorbed, and the thrust of a propeller from the characteristics of the sections employed in its construction as determined from wind tunnel tests. This method of analysis is usually referred to as the Drzwiecki method, although it was proposed independently by Lanchester at about the same

time that it was put forward by Drzwiecki. Figures 58 and 59 show a typical set of curves for a series of propeller sections as obtained from wind tunnel tests. The actual resultant pressure on the section may be obtained from the formula, $M = \rho$.

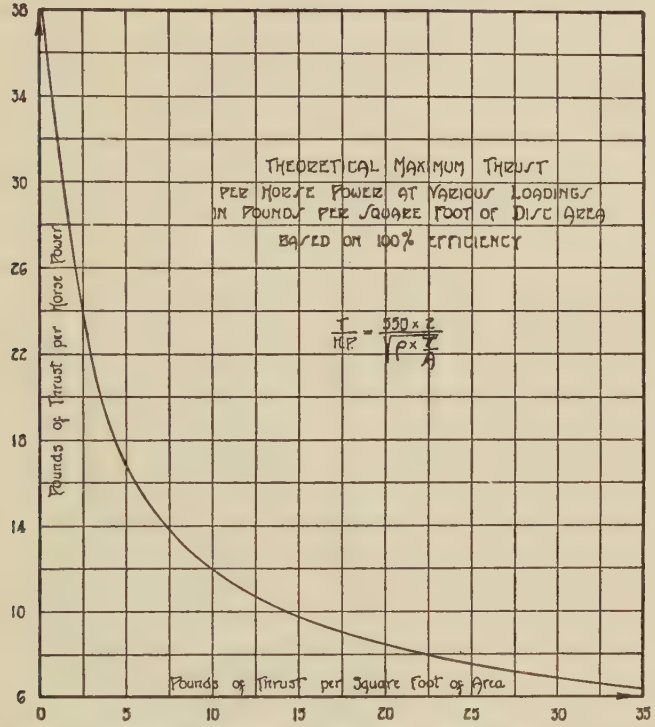


Figure 56. Theoretical Maximum Thrust per Horsepower at Various Loadings in Pounds per Square Foot of Disc Area Based on 100 Per Cent Efficiency

$K S V^2$, where M is the resultant pressure in pounds, K the resultant pressure coefficient, S the area of the airfoil in square feet.

In the study of the propeller however, it is necessary to know the direction of the resultant pressure as well as the magnitude. Figure 60 shows the angle between the normal to

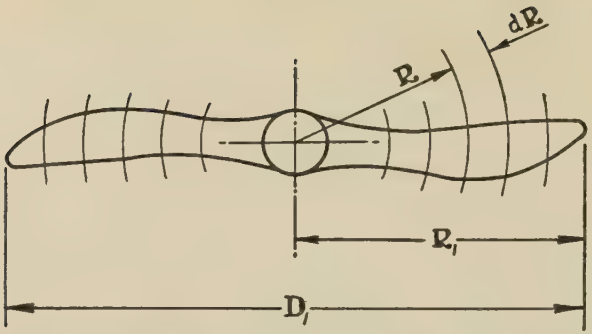


Figure 57. In applying the Drzwiecki Method of Analysis, the Propeller is Considered to be Made Up of a Number of Zones, Each Zone being a Small Airfoil, Acting in a Helical Path

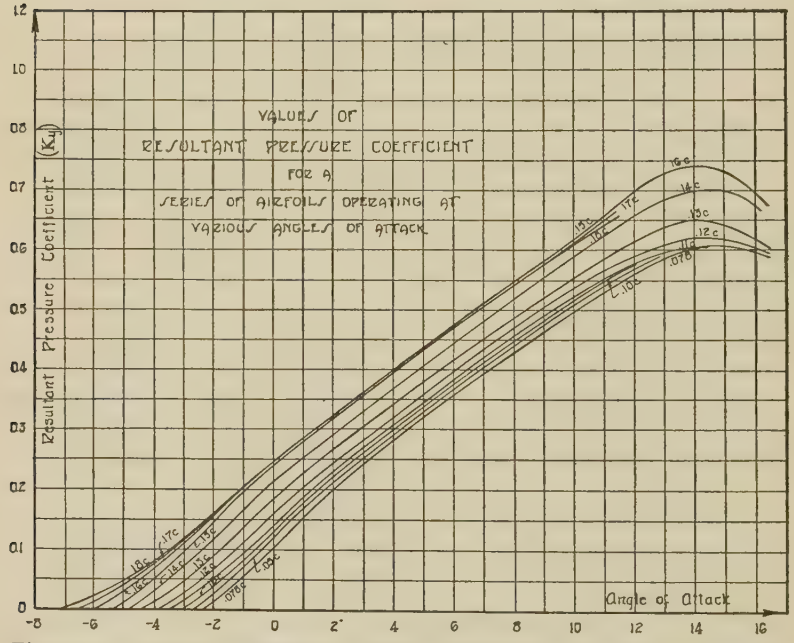


Figure 58. Values of Resultant Pressure Coefficient for a Series of Airfoils Operating at Various Angles of Attack

the path and the direction of the resultant pressure for the same series of propeller sections. This angle is designated as β.

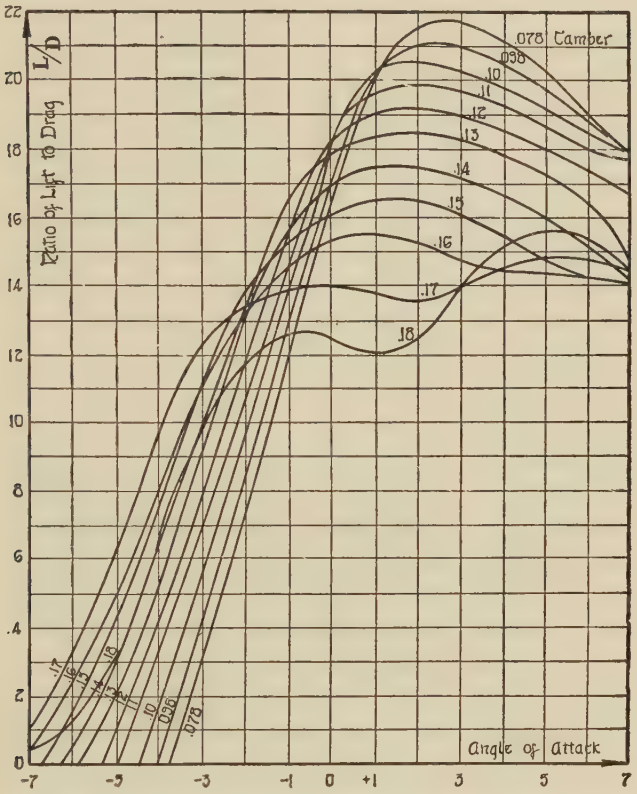


Figure 59. Ratio of Lift to Drag for a Series of Airfoils Operating at Various Angles of Attack

Figure 61 shows the velocity and force vectors for a typical zone of the propeller.

From an analysis of these forces, the following equation may be derived. The thrust on a small elemental strip

$$dT = \rho \left[\frac{\pi^3}{2} \cdot K \cdot \frac{b}{2\pi R} \cdot \left(\frac{R}{R_1} \right)^3 \cdot \frac{\cos(\phi + \beta)}{\cos^2 \phi} \cdot d\left(\frac{R}{R_1} \right) \right] n^2 D^4$$

The power on the same small elemental strip in foot pounds per second is obtained from the following equation:

$$dP_a = \rho \left[\frac{\pi^4}{2} \cdot K \cdot \frac{b}{2\pi R} \cdot \left(\frac{R}{R_1} \right) \cdot \frac{^4\sin(\phi + \beta)}{\cos^2 \phi} \cdot d\left(\frac{R}{R_1} \right) \right] n^3 D^5$$

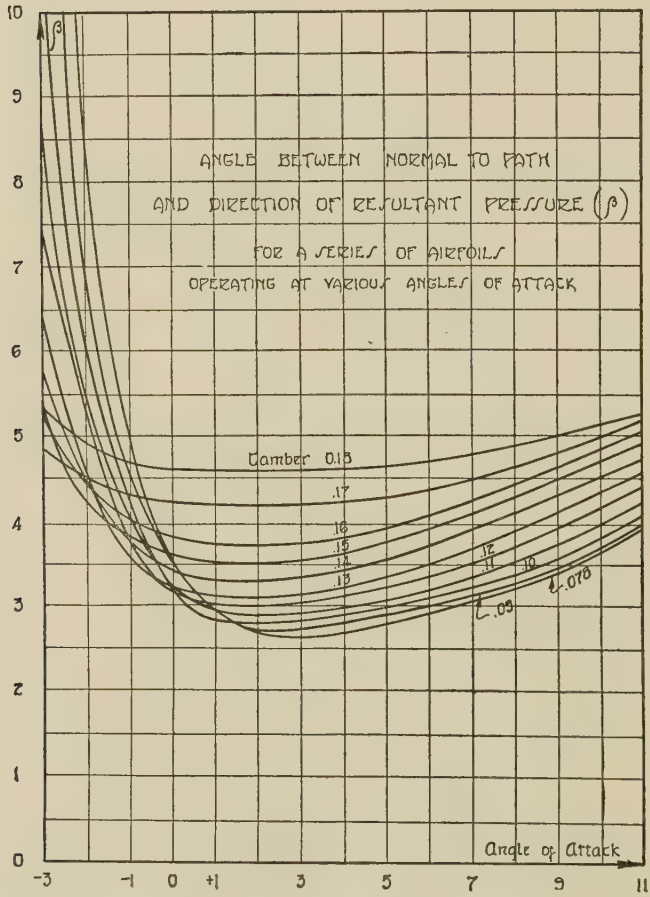


Figure 60. Angle Bewteen Normal to Path and Direction of Resultant Pressure (B) for a Series of Airfoils Operating at Various Angles of Attack

where b is the width of the propeller blade, R is the radius of the zone considered, R_1 the radius at the tip, the angles ϕ and β

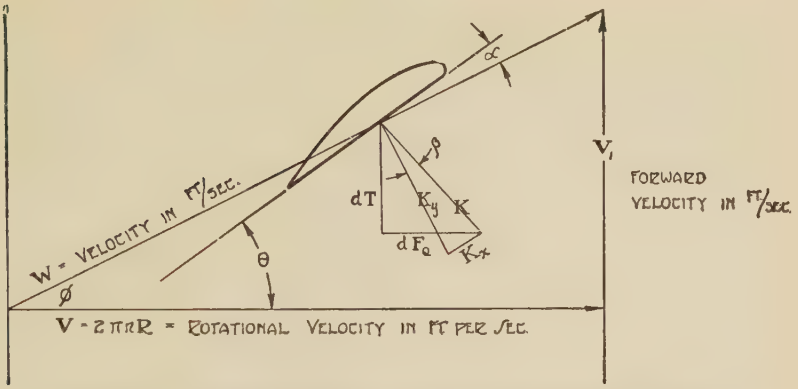


Figure 61. Velocity and Force Vectors for a Zone of the Propeller

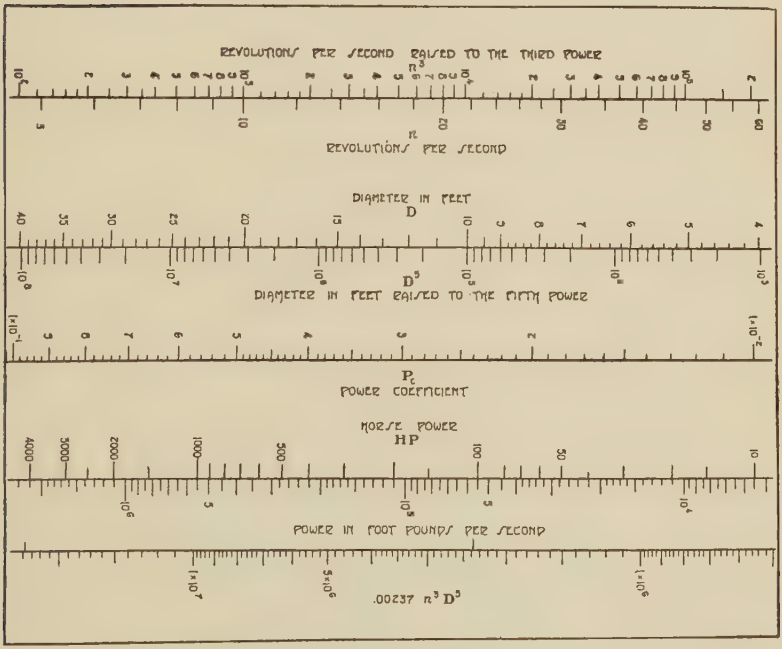


Figure 62. Nomogram for Solving the Formula $P = \rho\, P_c \eta^3 D^5$

those shown in the figure, n the revolutions per second, D the diameter of the propeller.

The integration of the quantity in the brackets gives the value for the thrust coefficient T_c and the power coefficient P_c.

The equations for the thrust and power of the whole propeller are usually written as follows:

$$T = \rho \cdot T_c \, n^2 D^4$$

The equation for the power is:

$$P = \rho \cdot P_c \, n^3 D^5$$

This value of the power is the horsepower $\times$ 550.

If we consider the work delivered by a small elemental strip of the propeller we must multiply the thrust by V_1 while the power absorbed is shown directly in the equation for power.

If we divide the work delivered by the power absorbed we find the airfoil efficiency of this particular section of the propeller and obtain the following equation.

$$e_2 = \frac{\tan \phi}{\tan (\phi + \beta)}$$

Obviously the efficiency as obtained from the airfoil method becomes greater with lower value for the angle β that is, the more nearly the resultant pressure is normal to the direction of motion.

The efficiency also varies with the angle ϕ and may be shown to be a maximum when, $\phi - \beta/2 = 45°$.

This expression for efficiency represents rather closely the percentage of the theoretical efficiency which may be obtained excepting that there is an additional loss due to the race rotation or rotational component of the slip stream. The product of the theoretical efficiency by this value, however, will give a very good approximation to the exact efficiency which may be obtained from the propeller.

Further developments of propeller theory. The foregoing discussion is meant to serve only as a brief elementary introduction to the propeller theory. The actual theory how-

ever involves a calculation of the inflow velocity which modifies the angle of attack of the propeller. At the same time it is necessary to correct the wind tunnel coefficients obtained from the propeller sections for infinite aspect ratio and to scale up the results of model tests for size and speed effects.

Static thrust of propeller. In applying the airfoil method to propellers for analysis of the static thrust and power absorbed when running with the aircraft at rest, it is necessary to take into account the inflow velocity of the slip stream as well as the race rotation or spin of the slip stream.

The angle of attack at which the propeller section strikes the air is modified by the inflow velocity and the race rotation so that it is first necessary to find the angle of attack in applying the airfoil method to conditions of static thrust. The following equation is used for this purpose:

$$\frac{2 \sin^2 \phi}{\cos(\phi + \beta)} = K \cdot \frac{B}{2 \pi R}$$

In this case ϕ represents the angle between the direction of the motion of the propeller blade with respect to the air and the plane of rotation, B the angle between the normal to the path and the resultant pressure, K the resultant pressure coefficient, B the sum of the width of all the blades at the station under consideration, R the radius of the station under consideration and the quantity $\frac{B}{2 \pi R}$ the breadth ratio or the ratio of total blade width to circumference at the station under consideration.

The angle of attack is the difference between the angle ϕ and blade angle. For the solution of this equation nomograms are given in Figures 63 and 64. In designing a propeller for static thrust it is often desirable to know the correct angle for the blade when the breadth ratio and angle of attack have been selected. For this purpose the resultant pressure coefficient and the angle β may be found from the wind tunnel coefficients of the propeller section and we may then use the

nomogram to solve for ϕ. The blade angle would then be equal to the angle ϕ plus the angle of attack.

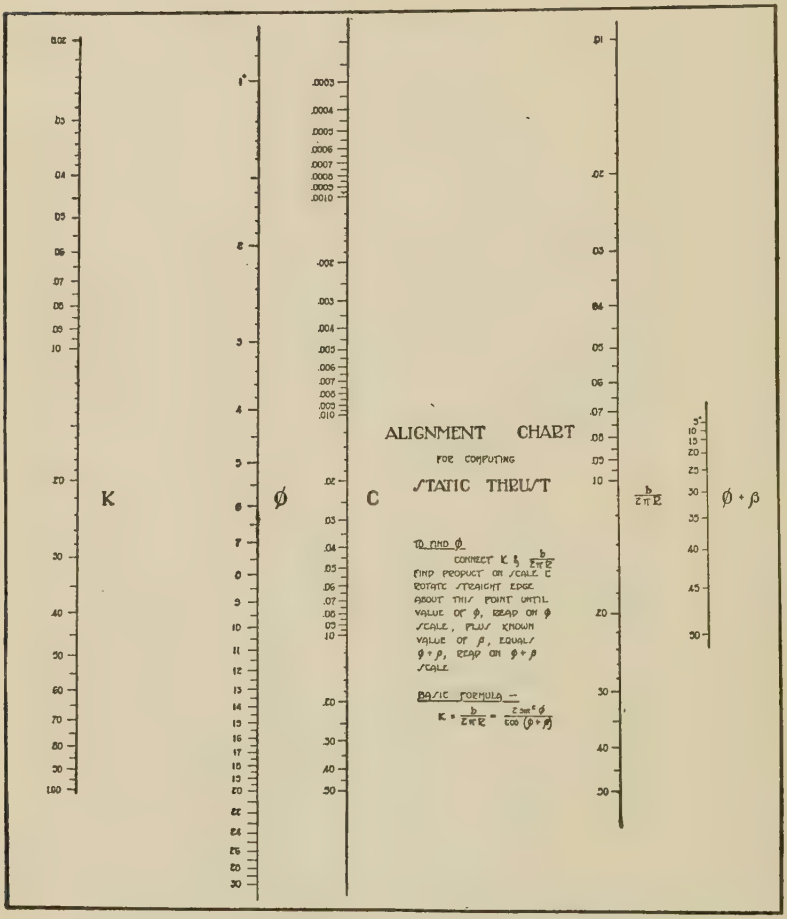


Figure 63. Alignment Chart for Computing Static Thrust

After K, ϕ and β and $\dfrac{B}{2\pi R}$ are known, the thrust coefficient power, and power coefficient may be obtained from the following equation:

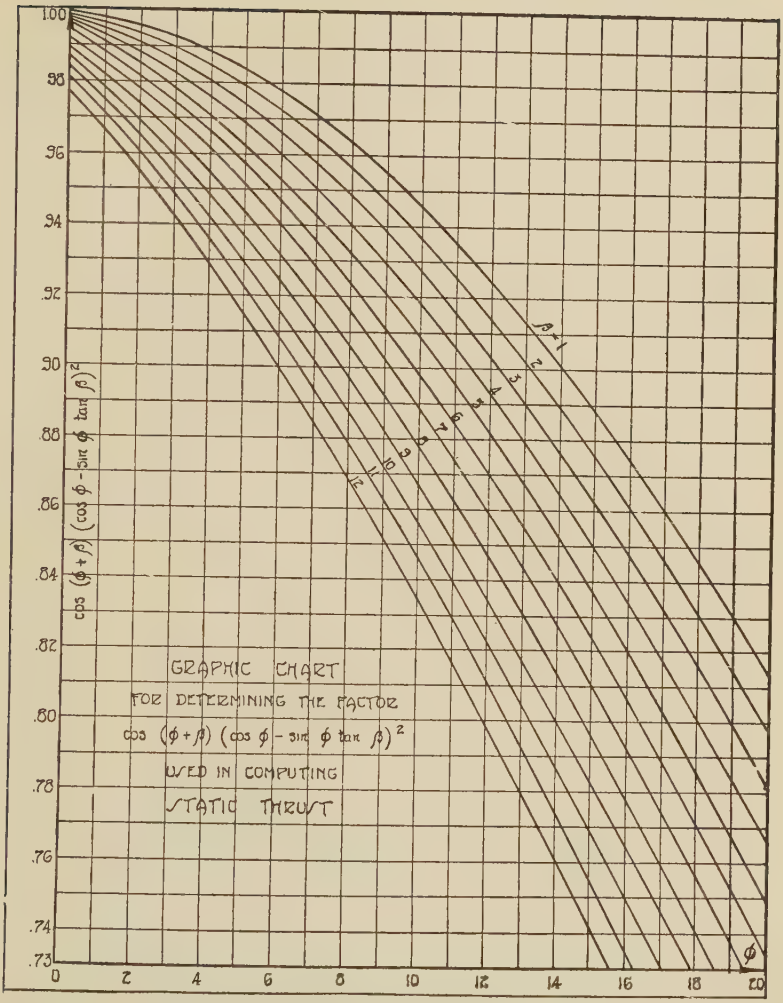


Figure 64. Graphic Chart for Determining the Factor $\cos(\phi+\beta)(\cos\phi-\sin\phi\tan\beta)^2$ Used in Computing Static Thrust

$$dT = \rho\left[\frac{\pi^3}{2}K\cdot\frac{B}{2\pi R}\cdot\left(\frac{R}{R_1}\right)^3(\cos\phi-\sin\phi\tan\beta)^2\cos(\phi+\beta)\cdot d\left(\frac{R}{R_1}\right)\right]n^2D^4$$

$$T_c = \int_0^{R_1}\frac{\pi^3}{2}K\cdot\frac{B}{2\pi R}\cdot\left(\frac{R}{R_1}\right)^3\cdot(\cos\phi-\sin\phi\tan\beta)^2\cos(\phi+\beta)\cdot d\left(\frac{R}{R_1}\right)$$

$$dP = \rho\left[\frac{\pi^4}{2}K\cdot\frac{B}{2\pi R}\cdot\left(\frac{R}{R_1}\right)^4\cdot(\cos\phi-\sin\phi\tan\beta)^2\sin(\phi+\beta)\,d\left(\frac{R}{R_1}\right)\right]n^3D^5$$

$$P_c = \int_0^{R_1}\frac{\pi^4}{2}K\cdot\frac{B}{2\pi R}\cdot\left(\frac{R}{R_1}\right)^4(\cos\phi-\sin\phi\tan\beta)^2\sin(\phi+\beta)\,d\left(\frac{R}{R_1}\right)$$

Wind Tunnel Tests of Model Propellers

In the wind tunnel tests of model propellers the tests are usually made on a model one-third to one-quarter the size of the full scale propeller and the result scaled up to apply to the full size propellers by means of the equations,

$$T = \rho\cdot T_c\, n^2D^4$$
$$P = \rho\cdot P_c\, n^3D^5$$

The wind tunnel efficiency however may be applied directly with corrections only for scale effects and for body interference.

Efficiency of the propellers as obtained from wind tunnel tests. Figure 66 shows a typical set of efficiency curves for a series of propellers designed for different pitch ratios and tested at various ratios of forward velocity to peripheral velocity of the propeller tip.

These curves show that the efficiency continues to increase until the ratio of the forward velocity to peripheral velocity is equal to about 0.4. If the curves shown extended further however, the efficiency obtained would again begin to decrease and would be zero when the ratio of forward to peripheral velocity is infinite.

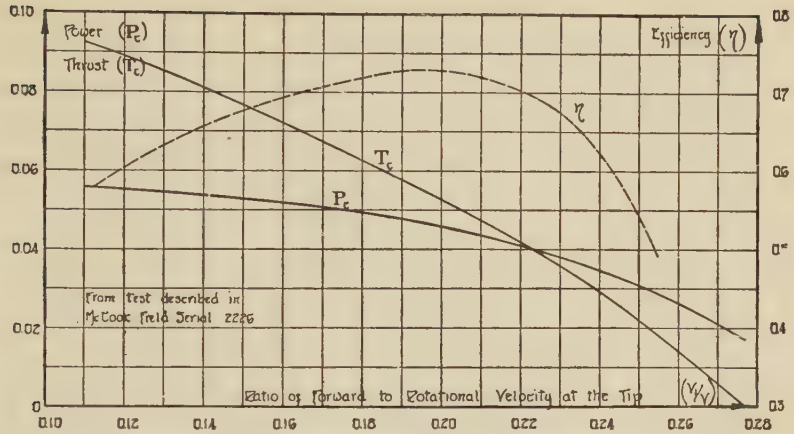


Figure 65. Typical Curves of P_c, T_c, and Efficiency, η

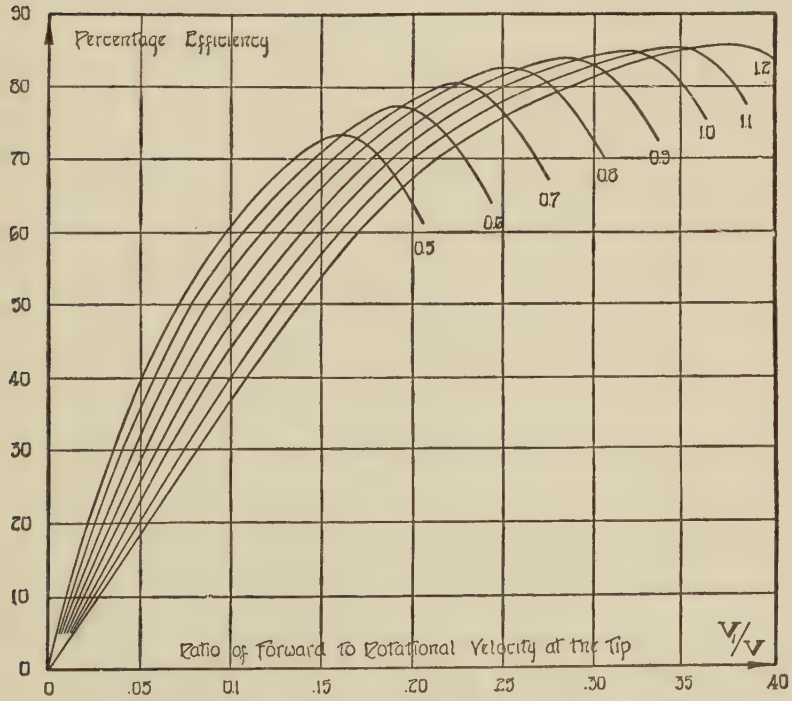


Figure 66. Typical Set of Efficiency Curves for a Series of Propellers

Ideal conditions for propeller efficiency. From these curves it may be seen that the propeller efficiency is greatest when the peripheral velocity at the tip is about two and one-half times the forward velocity of the aircraft. In the practical application of direct drive engines, the peripheral velocity is always much higher than this ideal value for airship service; frequently a gear reduction is used so that the propeller conditions may more nearly approach the ideal.

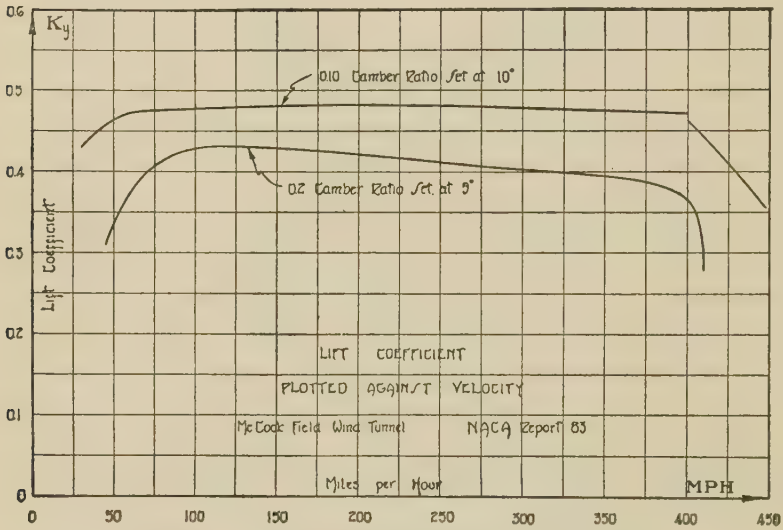


Figure 67. Lift Coefficient Plotted Against Velocity

The efficiency curves shown were run at a moderate speed and the values are rather high, representing the best to be expected from a series of propellers.

Effect of peripheral velocity on the efficiency of the propeller as obtained from wind tunnel tests. Figure 68 shows the effect of speed upon the resultant pressure coefficient for a fairly thin and a rather thick propeller section as obtained in the high-speed wind tunnel at McCook Field. In each case it may be noted that a very high velocity is reached where the

resultant pressure coefficient suddenly begins to decrease. At this same speed the angle β suddenly increases, both effects being due to a change in the type of flow occuring past the propeller section. Since an increase in the angle β means a considerable loss of efficiency, it is to be expected that propellers run at extremely high peripheral speeds would show consider-

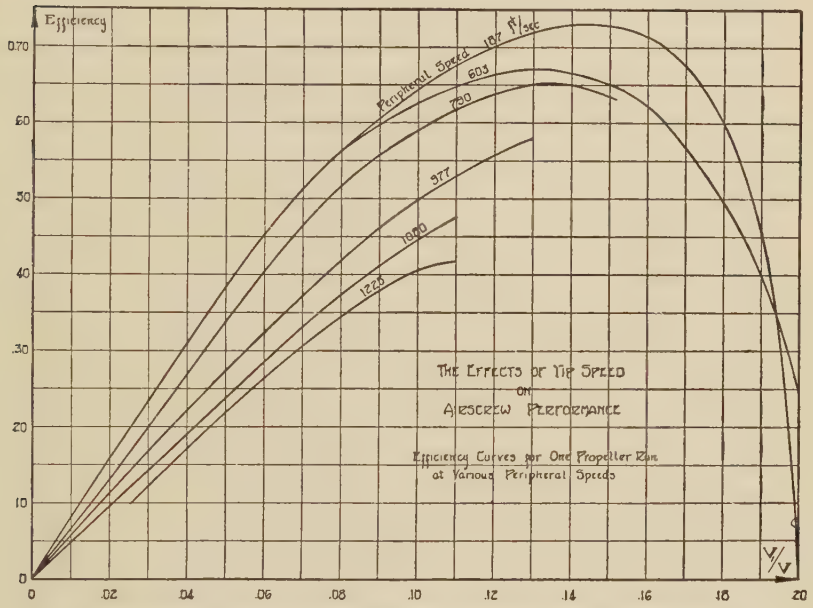


Figure 68. The Effects of Tip Speed on Airscrew Performance

ably lower efficiency than those run at moderate peripheral speeds.[1]

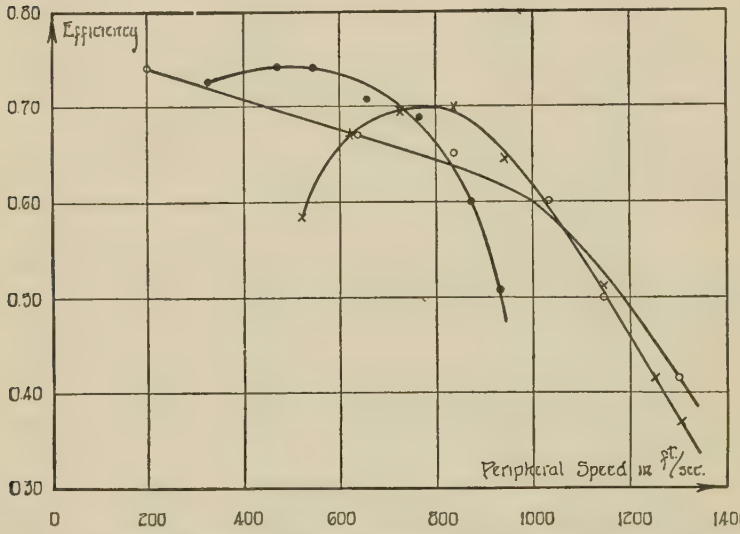
The effect of tip speed upon the efficiency of the propeller has been investigated by G. B. Douglas and R. McKinnon Wood.[2] The result of these tests are shown in Figure 69. It will be noted that the efficiency of the propeller that was tested, dropped from about 73 per cent at 187 feet per second

[1] See National Advisory Committee Report No. 83, "Wind Tunnel Studies in Aerodynamic Phenomena at High Speeds," by F. W. Caldwell and E. N. Fales.

[2] "The Effects of Tip Speed on Airscrew Performance," R & M Report No. 884, British Aeronautical Research Committee.

peripheral speed to about 42 per cent at 1,225 feet per second peripheral speed.

This is a very serious loss in efficiency and indicates that it may not be possible to use extremely high peripheral speeds efficiently. Fortunately however, the wind tunnel tests indicate



o — o — o Maximum efficiency of wooden propeller in R & M-884, Figure 5, at various peripheral speeds

● — ● — ● Efficiency of 10 feet 4 inches wooden propeller on static test, D. W. T., 367 at various peripheral speeds

× — × — × Efficiency of 10 feet duralumin propeller on static test, D. W. T. 620, at various peripheral speeds.

Figure 69. Efficiency of Two Propellers as Static Thrust Producers at Various Peripheral Speeds Compared with Efficiency of a Propeller in Flight at Various Peripheral Speeds

that the drop in efficiency for these sections occurs at a very much higher velocity on thin sections than on thick ones, consequently it is probable that the new thin metal propellers can be run to much higher peripheral speeds than has been possible with the relatively thick wooden propellers.

Comparison of wind tunnel tests of model propellers with the characteristics computed by the airfoil method. Figure

71 shows a comparison of the wind tunnel tests made by Dr. Durand at Stanford University on a series of model propellers designed by the writer with values of the power coefficient as computed from the equation;

$$dP_a = \rho \cdot \left[\frac{\pi^4}{2} K \cdot \frac{b}{2\pi R} \cdot \left(\frac{R}{R_1}\right)^4 \cdot \frac{\sin(\phi+\beta)}{\cos^2\phi}\, d\left(\frac{R}{R_1}\right)\right] n^3 D^5$$

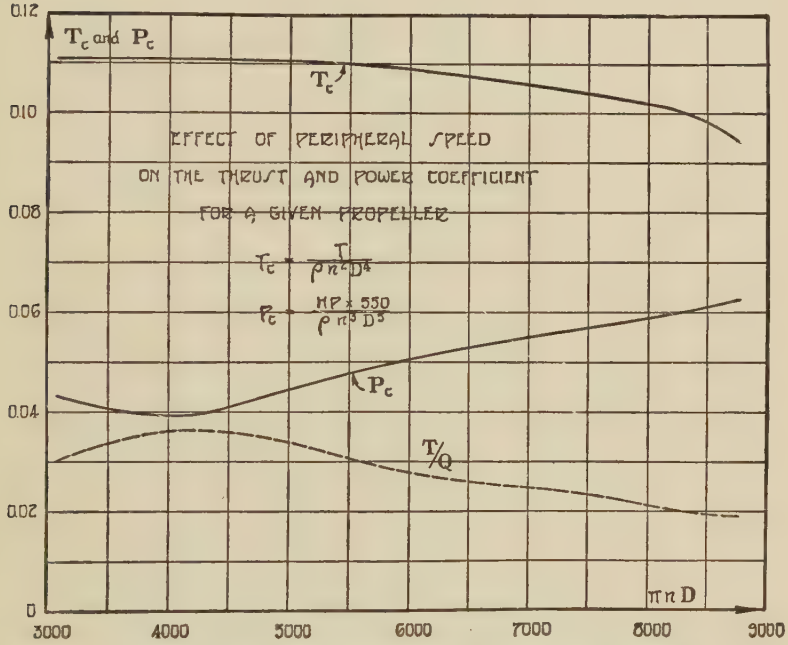


Figure 70. Effect of Peripheral Speed on the Thrust and Power Coefficient for a Given Propeller

These tests indicate that a systematic series of corrections may be applied to the computed values in order to obtain a check with the model tests.

Effect of Body Interference on Propeller Characteristics

In general a body placed in front of the propeller or behind it increases the power absorbed by the propeller at a given

speed of revolution, but also increases the thrust produced at the same time. The thrust is increased in a greater proportion than the power absorbed, so that the apparent efficiency of a propeller under these conditions is greater than that of the propeller in free air.

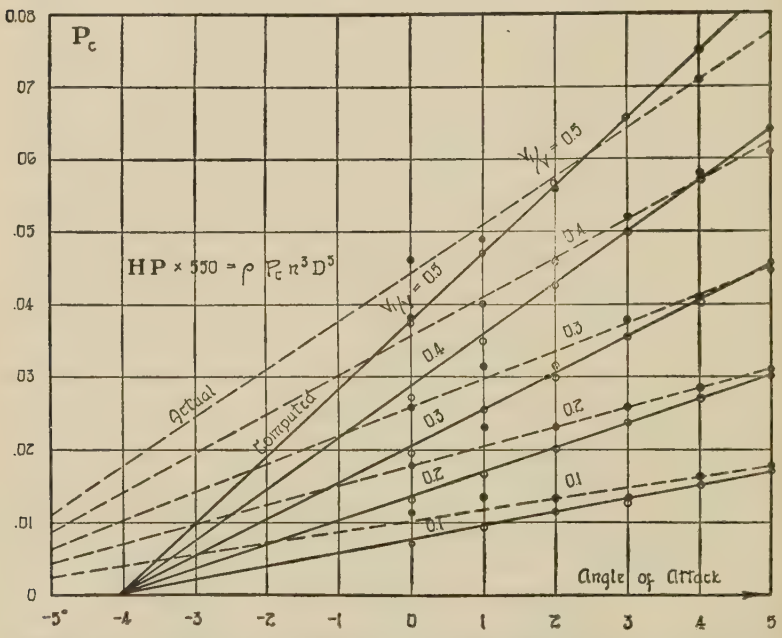


Figure 71. Comparison of the Values of the Power Coefficient for a Series of Propellers made from Planform 3, as obtained from calculations by the airfoil method with those obtained from actual model tests

However, at the same time that the thrust of the propeller is increased, the resistance of the body is increased due to its being placed near the propeller and the increase in body resistance is usually greater than the increase of propeller thrust so that the net result is usually a loss in efficiency instead of a gain.

Some tests made with perfect stream line bodies behind the propeller have shown an improvement in net propeller effi-

ciency of 2 per cent or 3 per cent. In these cases the increase in body resistance is not as great as the increase in thrust so that there is a net gain. In nearly all practical cases, however,

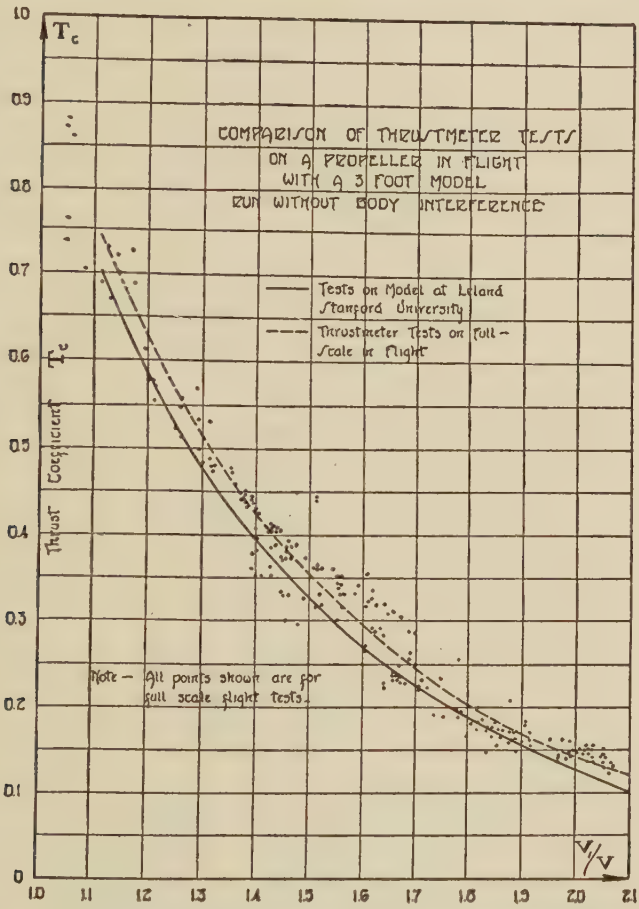


Figure 72. Comparison of Thrustmeter Tests on a Propeller in Flight with a Three-Foot Model Run Without Body Interference

the effect of the mutual interference is a very considerable loss in efficiency which varies from about 5 to 20 per cent according to the size, shape and position of the body. Figure 73,

from the British Aeronautical Research Committee,[3] shows the relative size of a series of pusher and tractor bodies used for interference tests between bodies and air screws.

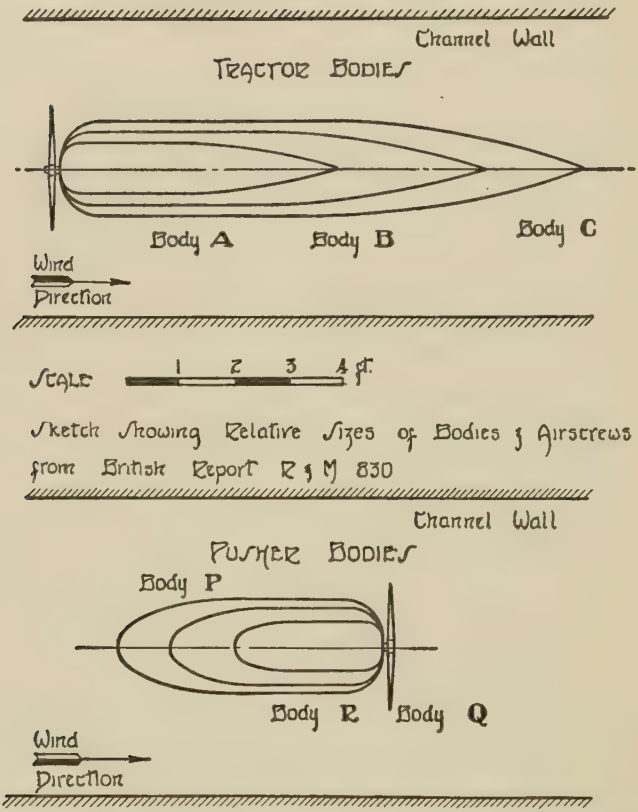


Figure 73. Tests of Mutual Interference of Bodies and Airscrews

The effect of the interference of these bodies on the net efficiency as obtained from the British tests is shown in Figure 74. In this case the tractor body is somewhat better than the pusher. This cannot be applied as a general rule to lighter-than-air craft, however, as many airships have a large number

[3] R & M No. 830, "Experiments on Airscrews with Tractor and Pusher Bodies."

of bracing wires, cables and other supports in the slip stream of the propeller, consequently the comparison for most airships would not be as favorable to the tractor arrangement.

It is not possible to draw specific conclusions as to the relative merits of tractor and pusher arrangements for airship propellers. In general however, it may be said that the tractor arrangement is probably better where the body housing the engine and propeller is nicely stream lined and relatively free

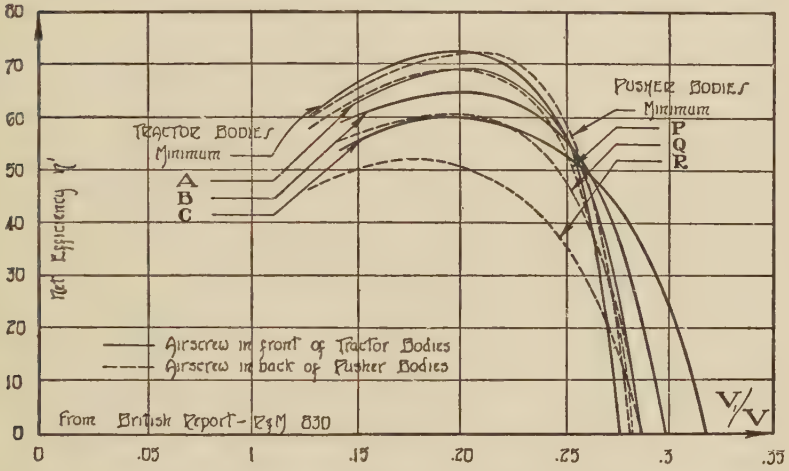


Figure 74. Net Efficiency $\left(\frac{(T - R)V}{2\pi nQ}\right)$ of a Four-Bladed Wooden Airscrew with Three Tractor Bodies, Three Pusher Bodies, and a Minimum Body

from exposed structural parts and supporting wires. In cases where the body is not well stream-lined and the supporting structure offers considerable resistance, it is possible that the pusher arrangement may be more efficient.

In addition to these British tests, some very interesting experiments on the effect of the fore-and-aft location of the body have been made by Professor E. P. Lesley at Leland Stanford University.

In general, these experiments seem to indicate that the fore-and-aft position of the body with respect to the propeller is

important and that it is desirable to have the propeller as far ahead of the body as possible in the tractor arrangement.

All of these experiments have been run with the body directly in line with the propeller axis. There are no experiments available on the effect of a very large body adjacent to the propeller as is the case in an airship. It is to be expected however, that the fore-and-aft location of the propeller along the ship will affect its characteristics owing to the variation in air velocity at various points along the ship.

CHAPTER X

PROPELLER DESIGN AND STRESS ANALYSES

Design of Propeller

The actual design of the propeller is usually one of adaptation. The top speed of the aircraft is fixed by its resistance and by the power available to drive it. The speed of revolution at which the propeller must turn is fixed within certain limits by the engine characteristics since the engine power at full throttle is about proportional to the r.p.m., and the engine must be allowed to turn at the proper r.p.m. if it is to develop the desired horsepower.

The propeller problem then becomes one of providing a propeller which will transform a given brake horsepower of the engine at a given r.p.m. into a maximum thrust horsepower at a given forward speed of the aircraft.

In the case of the airship the speed of the ship is nearly proportional to the r.p.m. of the propeller so that the ratio of forward to peripheral speed is constant provided we have only one propeller, or provided all the propellers are reduced in speed by the same percentage.

The efficiency of the propeller is dependent only on the ratio of forward to peripheral velocity so that the propeller which is best for top speed will also be equally efficient for cruising at reduced throttle.

In the actual design of a propeller, the designer starts with the horsepower of the engine, the r.p.m. of the propeller, and the speed of the aircraft.

The first step in the design is to determine the best propeller diameter. For this purpose the nomogram shown in Figure 75 may be used. This nomogram is based on British experiments

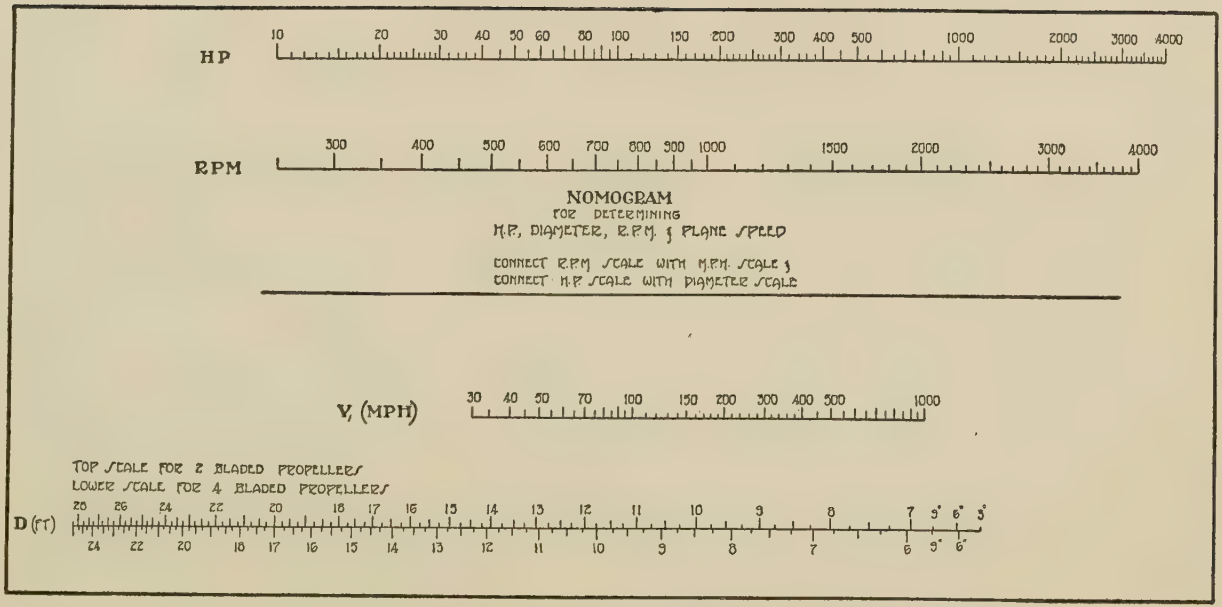


Figure 75. Nomogram for Determining Hp., Diameter, R. P. M., and Plane Speed

with planform No. 3 using a ratio of tip radius to maximum blade width of 6. This ratio is usually referred to as the aspect ratio.

Having determined the diameter and the aspect ratio, it is then necessary to choose the correct angle of attack for the propeller sections.

The equation for the propeller horsepower is given by the formula: hp. $\times 550 = \rho P_c n^3 D^5$. The monogram shown in Figure 62 (page 127) may be used to solve the equation and obtain P_c when the horsepower, diameter and r.p.m. are known.

The value of P_c obtained from this nomogram will be the value of P_c for the whole propeller. As there is always more than one blade, it is necessary to divide this value by the number of blades to determine the power coefficient, P_c per blade.

Knowing the power coefficient for the propeller, it is necessary to find the ratio of the forward to the peripheral velocity before we can determine the correct value for the angle of attack.

The forward velocity is the top speed of the aircraft and is expressed in feet per second for the purposes of propeller calculation. The peripheral velocity is also expressed in feet per second and is equal to πnD, where n is the revolutions per second.

Knowing the value of the power coefficient per blade and the ratio of forward to peripheral velocity, we may now obtain from the test values, shown in Figure 71, the correct value for the angle of attack corresponding to this power coefficient and ratio of forward to peripheral velocity.

The angle of advance, or effective pitch angle at the various stations, is designated as ϕ. The value of ϕ may be obtained from the formula:

$$\tan \phi = \frac{V_1}{2\pi nR}$$

where V_1 is the forward speed of the airship, n the revolutions per second, and R the radius at the station under consideration.

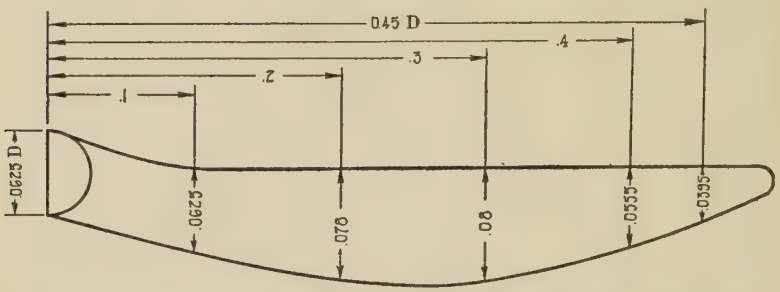


Figure 76. Planform of Micarta Liberty and 300-Hp. Wright Propellers

After obtaining the value of angle of advance at the various stations, the angle of attack is added to give the correct blade angle.

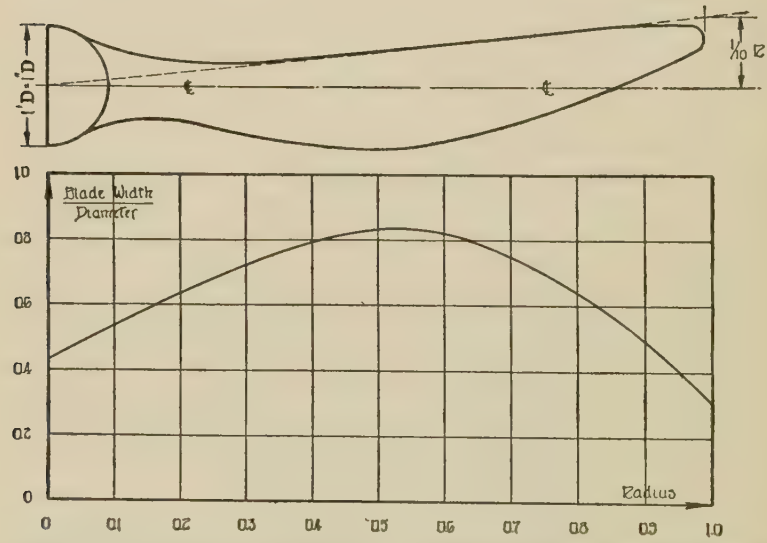


Figure 77. Planform of Blade No. 1. (Originated by the British Air Board)

The maximum blade width may be taken as one-sixth of the tip radius, while the blade width at the various stations

are obtained by multiplying the maximum blade width by the factor obtained from planform No. 3 shown in Figure 78.

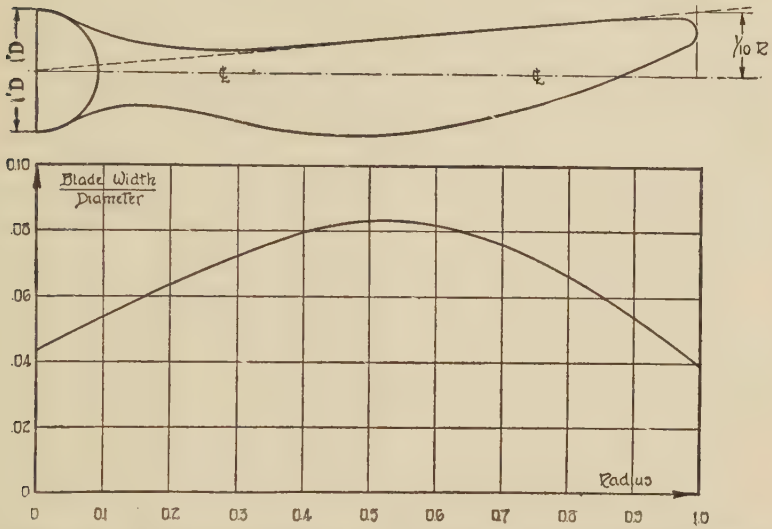


Figure 78. Planform of Blade No. 3. (Originated by British Air Board)

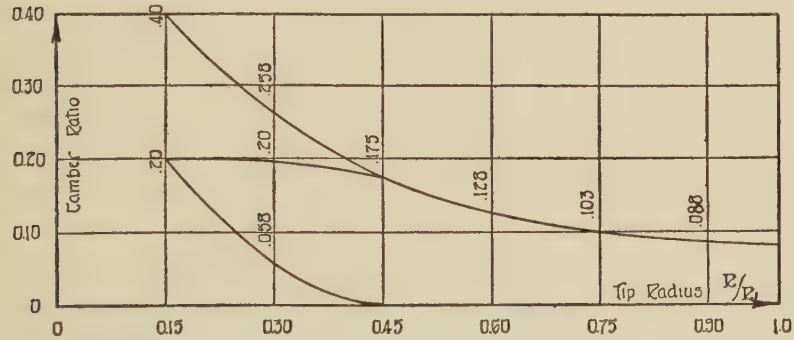
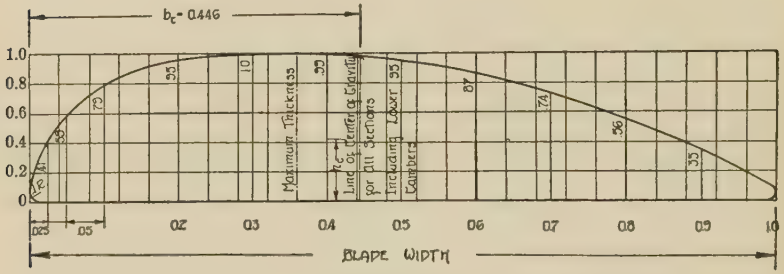


Figure 79. Camber Ratios Suitable for Use with Planforms 1 and 3 for Wooden Propellers

The maximum thickness at the various stations may be obtained from the curves of camber ratios shown in Figure 79. Knowing the blade width and the maximum thickness, the shape

of the cross-section is determined from the ordinates of the typical airfoil shown in Figure 80.



Area = 0.738 × b × t
C . G. above chord = 0.421 × t C. G. from leading edge = 0.4469 × b

Figure 80. Dimensions of Typical Section Suitable for Propeller Design

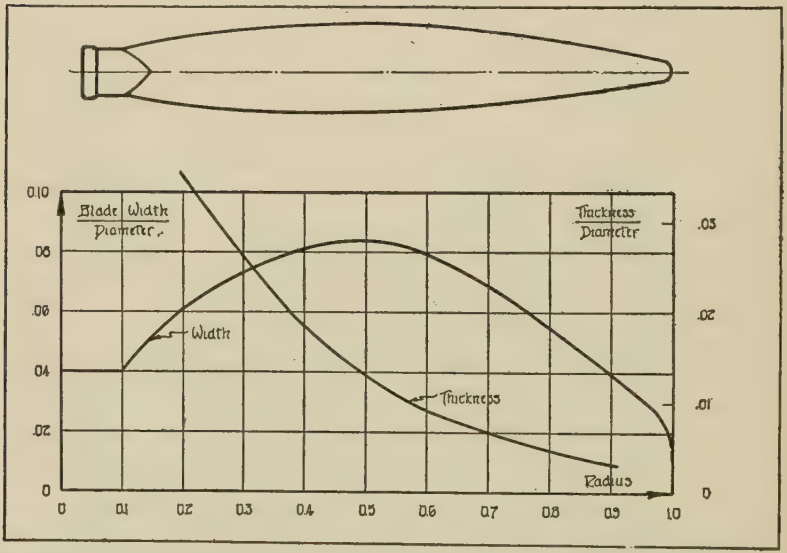


Figure 81. Planform for Micarta Propellers. (As used by the author)

The foregoing will give the basic information required to determine the propeller characteristics. The description is a typical method of design and will give a satisfactory wooden

propeller for tip speeds up to about 700 or 800 feet per second.

For special cases, other plan forms are more adaptable and it is often desirable to use a higher aspect ratio than 6; it is sometimes necessary to use a lower aspect ratio in case the possible diameter is very much lower than the best diameter.

For materials other than wood, a different plan form and thickness ratio will be necessary, as these dimensions must

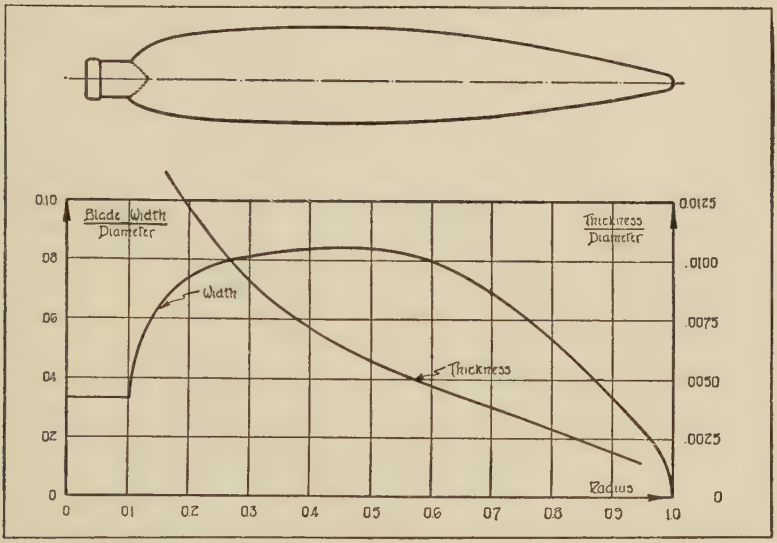


Figure 82. Planform for Adjustable Aluminum-Alloy Propellers. (As used by the author)

be determined from the stress analysis as well as from aerodynamic considerations; these depend, to a large extent, upon the mechanical properties of the material.

Figure 81 shows the planform and thickness ratio suitable for the design of micarta propellers. This planform and thickness ratio are satisfactory up to peripheral speeds of about 900 feet per second.

Figure 82 shows a planform and thickness ratio suitable for aluminum-alloy propellers. This planform and thickness

ratio are satisfactory up to peripheral speeds of about 1,200 feet per second.[1]

Rake of Propeller

Propellers are often given a forward tilt or rake as a means of balancing the bending moments due to air pressure by those due to centrifugal force. It is possible to eliminate the bending moments by this practice for any given ratio of forward to peripheral velocity and any given air density. It is not possible to eliminate the bending moments under all conditions however; where the practice of giving a forward rake is followed it is customary to design for some intermediate condition. Propellers are sometimes given a rake towards the trailing edge when viewed from the direction of the propeller axes. The object of this practice is to dampen out the torsional oscillations that produce flutter.

Stress Analysis of the Propeller

The actual stresses in the propeller are extremely complicated, owing to the distortion caused by the loads imposed and the effect of this distortion in modifying the stresses.

Static analyses. The static analysis as usually carried out is based upon the following loads:

1. Direct tension due to centrifugal force.
2. Bending moments in the thrust plane due to centrifugal force.
3. Bending moments in the torque plane due to centrifugal force.
4. Bending moments due to thrust.
5. Bending moments due to torque.

[1] For a more complete description of the design of aircraft propellers, see "Design of Screw Propellers for Aircraft," by H. W. Watts, and "The Airplane Propeller," War Department Document 998.

6. Bending moments due to the gyrostatical couple.
7. Torsion due to the air pressure reaction.
8. Torsion due to centrifugal force.

The *direct tension due to centrifugal force* is usually obtained by plotting a curve of centrifugal force per inch of radius against the radius in inches. Integration under this curve from the tip of the station under consideration gives the total centrifugal load on any section. This load divided by the area of the cross-section gives the direct centrifugal tension.

The bending moment on any station due to *centrifugal force in the thrust plane* is obtained by passing planes through the centers of gravity of all stations further out on the blade, these planes being normal to the axis of rotation and measuring the perpendicular distance from the center of gravity of the station under consideration to each of these planes. This gives the moment arm which, multiplied by the centrifugal force per inch of radius of the various outer stations, gives the bending moment per inch of radius on the station. Integration under this curve gives the total bending moment in the thrust plane on the station under consideration.

The *bending moment in the torque plane due to centrifugal force* is obtained by passing planes through the centers of gravity of all the stations further out on the blade than the station under consideration and through the axis of rotation. The perpendicular distance from the center of gravity of the particular station to these planes, gives the moment arm for the various outer stations, and the product of this moment arm by the centrifugal force per inch of radius of the various outer stations, gives the bending moment per inch of radius on the station being considered. Integration under this curve gives the total bending moment due to centrifugal force in the torque plane acting on the station under consideration.

In order to obtain the *bending moment due to thrust,* a curve of thrust per inch of radius is plotted against the radius. The

thrust per inch of radius at various outer stations is multiplied by the distance from these stations to the section under consideration to obtain the bending moment per inch of radius at the various outer stations. This gives a series of curves which, integrated, give the value of the total bending moment due to thrust on the various stations.

The *bending moments due to torque* are obtained by plotting a curve of torque force per inch of radius and proceeding as in the case of thrust.

The *bending moment due to the gyrostatical couple* may be obtained from the formula

$$M = \omega_1 \times \omega_2 \times I$$

where ω_1 is the angular velocity of the propeller in radians per second, ω_2 the angular velocity of the airplane in radians per second, and I the moment of inertia of the propeller in slugs feet squared. This bending moment in pounds-feet must be multiplied by 12 to find the bending moment in pounds-inches.

The *torsion due to air pressure reactions* is very difficult to determine on account of the lack of knowledge of the center of pressure location for propeller airfoils at very high speeds. It is of considerable importance in connection with the propeller stresses, however, as it frequently leads to a form of torsional vibration known as flutter.

The *torsional bending moment due to centrifugal force* is obtained by the method given under the heading of Adjustable Pitch Propellers, page 156.

The total bending moments in the thrust and torque planes may be obtained by the algebraic sum of the various bending moments, and the location of the most highly strained fibre may be found by the method of the neutral axis given in any good textbook on Mechanics.

The section properties for the typical airfoil section, shown

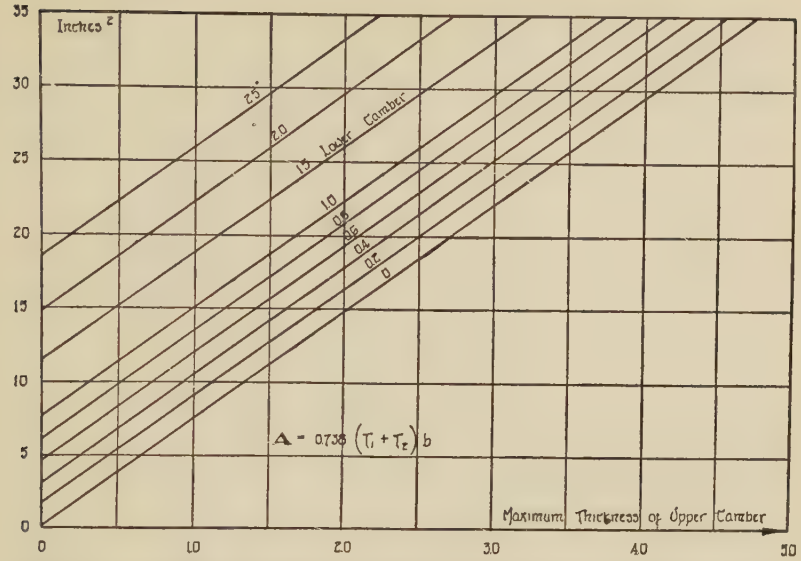


Figure 83. Section Properties of Standard Airfoils for Propellers. (Area of Section "A" for 10-inch blade width)

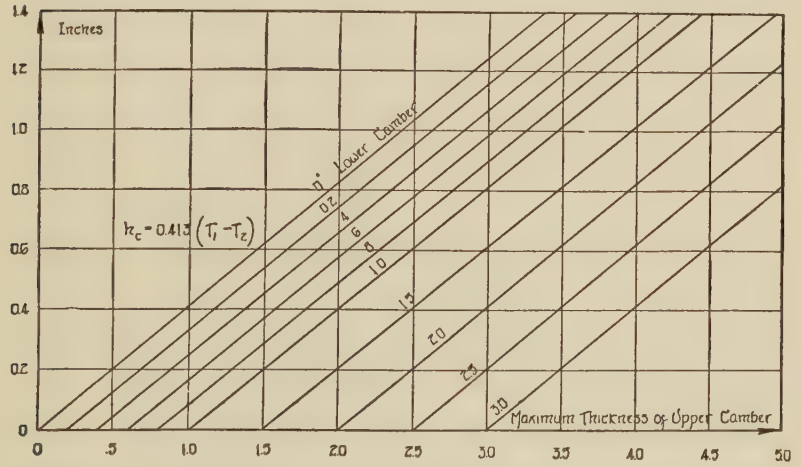


Figure 84. Distance C. G. Above Chord (b_c) for 10-inch Blade Width

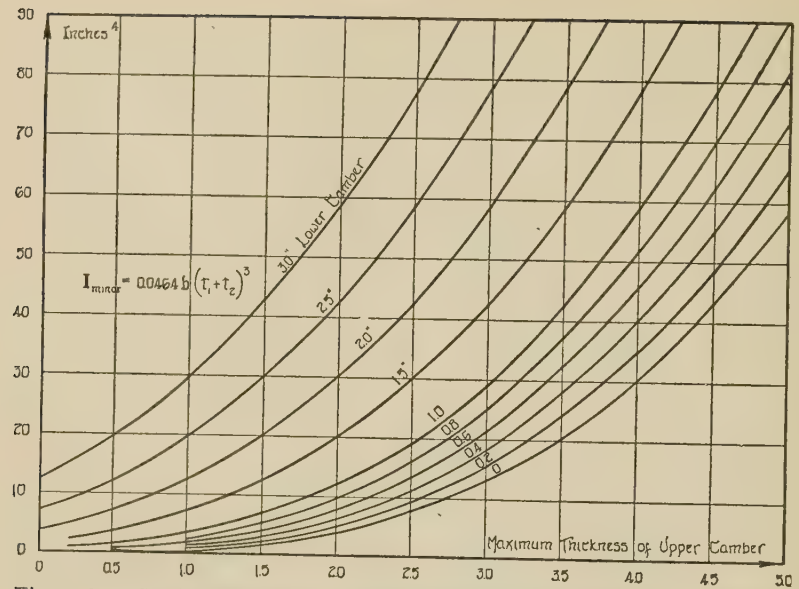


Figure 85. Section Properties for Standard Airfoil for Propellers. [Moment of Inertia (I_m) about Minor Principal Axis]

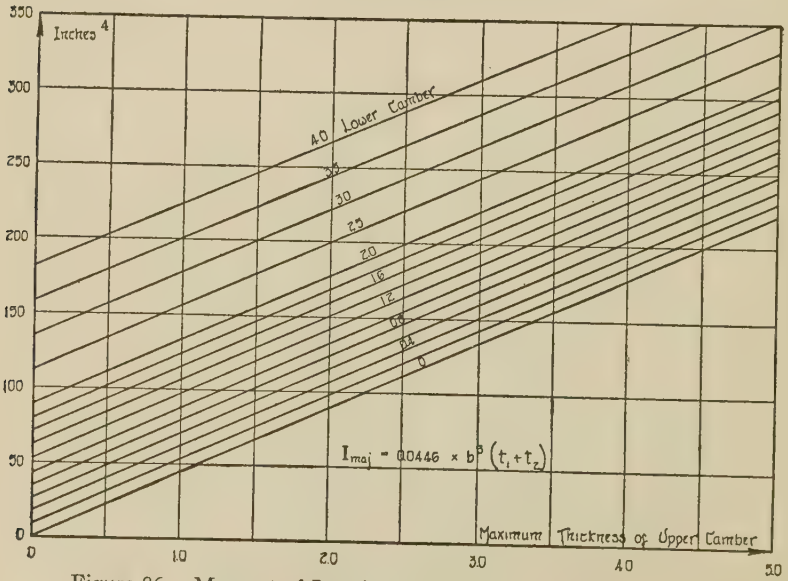


Figure 86. Moment of Inertia (I_m) about Major Principal Axis

in Figure 80, have been worked out in some detail in Figures 83 to 87 inclusive. By the use of this data the section properties may be determined very rapidly and with a considerable degree of accuracy.[2]

Flutter of propellers. The flutter of propellers is a form of torsional oscillation. It is supposed to be set up by the shifting of the center of pressure on the propeller blade and

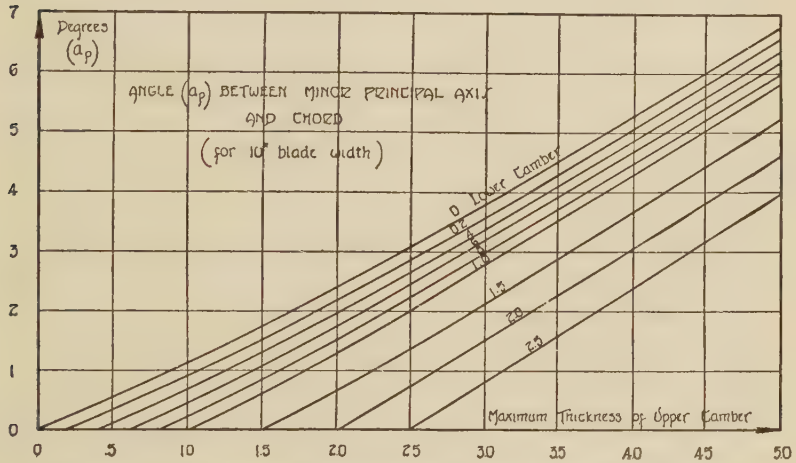


Figure 87. Angle (a_p) between Minor Principal Axis and Chord

is often an unstable oscillation which increases in amplitude until the propeller is destroyed. It can be recognized by the sharp explosive sounds which are produced when the propeller starts fluttering. Flutter is more often found in the airplane propellers than on the airship propellers, since the peripheral velocity used in airship propellers is usually lower than that used for the airplane propellers. Flutter is apparently affected by the planform of the blades and can often be eliminated by the use of a slight trailing rake of the propeller blade.

[2] For more complete description of the method of stress analysis of propellers, see "The Airplane Propeller," War Department Document No. 998, or "The Design of Screw Propellers for Aircraft," by H. C. Watts.

Adjustable Pitch and Reversible Propellers

The reversible propeller is of considerable interest for airship service as some form of reversing gear must be applied in large airships.

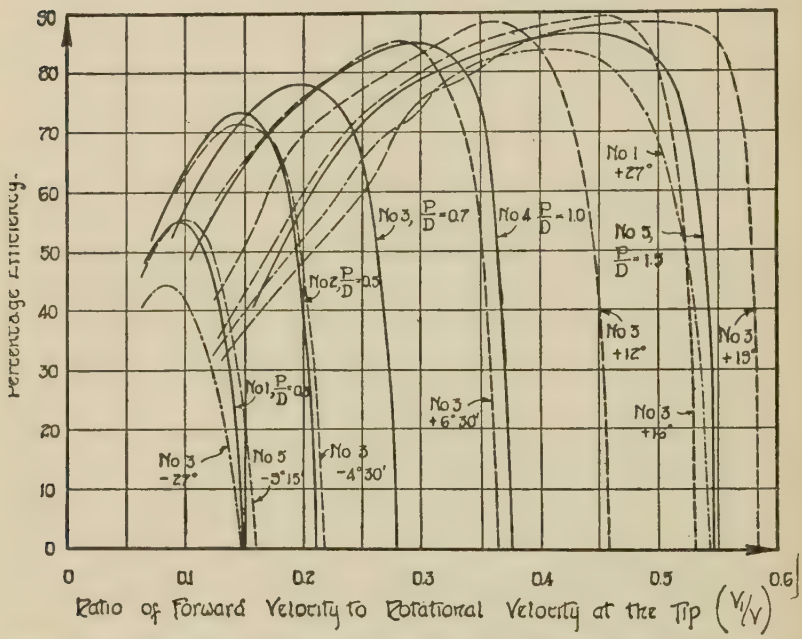


Figure 88. Tests with a Series of Two-Blade Adjustable Pitch Propellers Having a Maximum Blade Width of 0.082 Diameter

The reverse thrust from the propeller may be obtained from one of three ways:

1. By reversing the direction of rotation of the engine by means of special camshaft attachments.
2. By reversing the directions of rotation of the propeller alone through the action of intermediate gearing.
3. By changes in the pitch angles of the propeller, such that its thrust is reversed in direction.

A considerable amount of experimenting has been done in the development of a satisfactory reversible propeller. Most of

the work has been done with a view to its application to airplanes. In the case of the airplane application, it must be possible for the pilot to shift the pitch of the propeller at full throttle without undue effort and must also be possible for him to reverse the propeller completely in a time less than one second. These requirements do not apply to airships to such an extent, so that the problem of reversible propellers for lighter-than-air craft is much more simple than for the airplane.

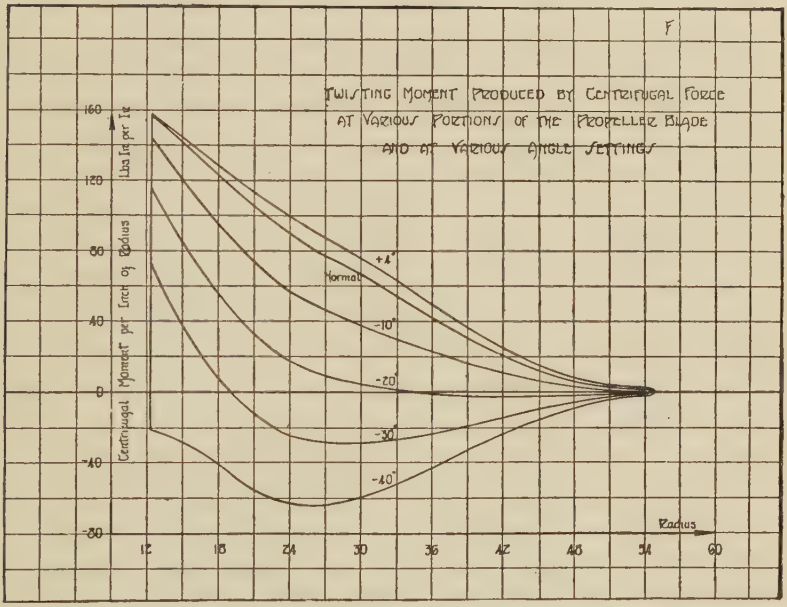


Figure 89. Twisting Moment Produced by Centrifugal Force at Various Portions of the Propeller Blade and at Various Angle Settings

A question of considerable interest in connection with adjustable pitch propellers is the effect of adjusting the pitch on the efficiency of the propeller. Figure 88 shows the effect of adjusting the pitch on the efficiency as obtained from a series of British tests in the wind tunnel. These tests indicate that there is no loss of efficiency due to the adjustment of pitch.

In the design and operation of reversible propellers, it is desirable to be familiar with the various twisting moments which tend to alter the pitch of the propeller opposing the action of the control.

The most important of these twisting moments is introduced by centrifugal force which can be shown to be:

$$Q_R = \frac{\delta}{g}\omega^2 \int (I \text{ major} - I \text{ minor}) \sin \alpha \cos \alpha \, dR$$

where δ is the density of the material, g the acceleration due to gravity, ω the rotational velocity in radians per second, I major and I minor, the moments of inertia of a section of the propeller about the principal axes, α the angle between the minor axes of the section of the propeller and the plane of rotation.[3]

Figure 90 shows the twisting moment computed in this manner for a small propeller at the various pitch settings. It may be seen that this twisting moment becomes zero for one setting. It may also be shown that the twisting moment would be zero again at another setting at 90° to the one shown. This second setting however, is outside of the range of pitch adjustment in actual use, but is of interest since it is possible to apply a counter weight at 90° to the setting of zero twisting moment, and this counter weight will exactly balance the centrifugal twisting moment of propeller at all pitch settings and speeds.

A second twisting moment is introduced by the eccentric location of the center of air pressure on the blades. This cannot be accurately estimated on account of the lack of knowledge of the center of pressure location for airfoils operating at very high speeds.

Considerable difficulty has been encountered in holding wooden blades in the metal hubs in the case of the wooden adjustable propellers. This trouble has been overcome in a satisfactory manner for engines up to 200 horsepower, but still presents much difficulty for larger engines.

[3] For more detailed information, see "American Progress in Adjustable Pitch Propellers," by F. W. Caldwell, International Congress of Aerial Navigation, London, England, 1923.

The introduction of the new aluminum alloy and micarta propellers has greatly simplified the reversible propeller problem so that it is now possible to build reversible propellers for almost any power desired.

One feature in connection with reversible propellers which requires particular care both in design and operation is the tend-

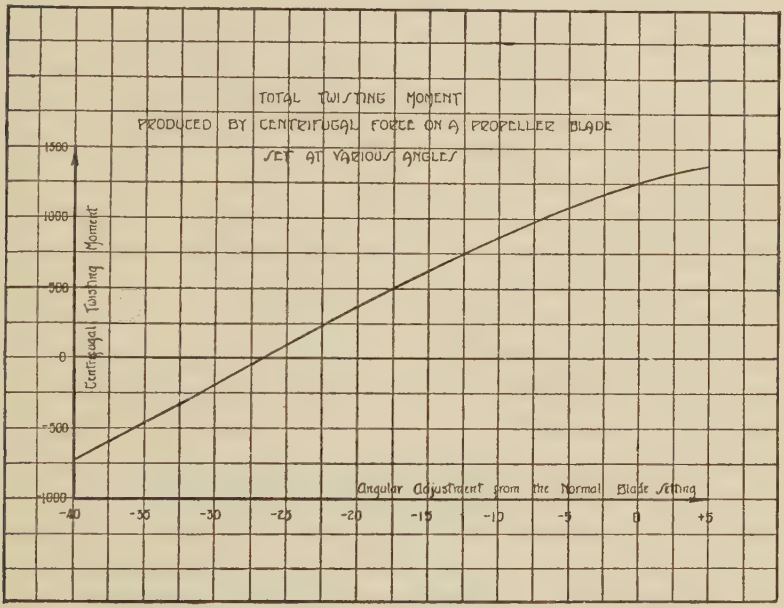


Figure 90. Total Twisting Moment Produced by Centrifugal Force on a Propeller Blade Set at Various Angles

ency of the engine to race with the propeller set in neutral pitch. If the propeller is set in neutral pitch the engine will turn at such a high speed as to destroy either the engine or propeller or possibly both. In order to avoid this possibility all reversible propellers should be equipped with a device for closing the throttle when the pitch of the propeller is in neutral position.[4]

[4] For a description of a number of types of reversible propellers, see "American Progress in Adjustable Pitch Propellers," by F. W. Caldwell, International Congress of Aerial Navigation, London, England, 1923.

CHAPTER XI

PROPELLER MATERIALS AND CONSTRUCTION

Wood Propellers

General considerations. In some respects wood is a very excellent material for propeller construction inasmuch as a number of woods have an extremely high tensile strength in proportion to their weight. In spite of the development of some splendid light alloys and high-tensile steels, birch wood is still superior to any material other than wood in terms of strength per unit of weight. In other words, among a number of pieces of one foot lengths made of various materials and all of such proportions that their weight is the same, a piece of birch will show a greater tensile strength than any of the other materials.

Wood however, is relatively weak in compression, and in spite of the tensile stress introduced by centrifugal force, certain parts of the propeller blades are sometimes subjected to relatively great compression stresses, so that the factor of safety in compression is sometimes lower than that in tension.

It must also be borne in mind that wood has very low shearing strength and is particularly weak in torsional shear. In addition to the stresses indicated by the usual methods of static analysis, the propeller is often subjected to very severe torsional stresses resulting from a form of torsional vibration known as fluttering. For this reason wood propellers often split under a comparatively mild overload, and if not properly designed this splitting may lead to tension or compression failures which wreck the propeller.

An additional disadvantage of wooden propellers is their tendency to warp and split due to the effect of climatic condi-

tions. This warping and splitting has been particularly noticeable in very dry climates such as New Mexico and Arizona.

In very moist climates the glued joints of wooden propellers are sometimes affected by the moisture to such an extent that the glue joints fail. This is particularly troublesome in tropical climates such as Panama and the Philippines.

Recent improvements in the methods of protection for propellers such as the aluminum leaf coating, similar to the sign painter's leaf, have been helpful in meeting these conditions, but have not entirely overcome the difficulties.

However, wooden propellers have the great advantage that they are much cheaper to build experimentally than any other type; they continue to be used largely for the reason that aircraft construction is still very much in the experimental stage.

Kiln drying of wood for propeller construction. It is very important that the lumber entering into propeller construction be carefully kiln dried so as to avoid initial strains in the material which might lead later to warping; also to provide a uniform moisture content for the wood throughout the propeller.

As is generally known, the contraction and expansion of wood vary rapidly with changes in moisture content although its contraction and expansion as the result of temperature changes alone is not very great.

Green lumber may contain about 50 per cent moisture and during the kiln drying process it is usually desirable to bring this moisture content down to about 5 per cent. The strains introduced in drying wood are the result of unequal contraction of various parts of the board, which in turn is caused by unequal rates of drying throughout the parts of the board. For this reason it is essential that the board should be dried uniformly from the surface to the center.

In order to accomplish this result it has been found necessary to carry out the drying process rather slowly so that the

rate of drying may be uniform throughout. The rate at which the drying is carried out depends on the relative humidity or percentage of saturation of the air surrounding the lumber. The drying process is usually begun with a very high degree of saturation, the percentage of saturation of the air being gradually reduced as the drying proceeds.

As a general relation between the humidity of the atmosphere and the percentage of moisture that lumber will contain if exposed indefinitely to that relative humidity, a curve (Figure 91) is given showing these relations graphically.[1]

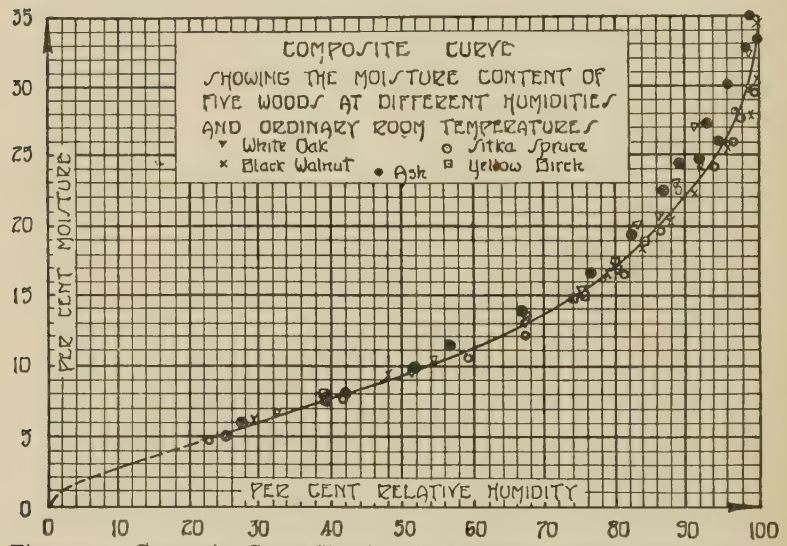


Figure 91. Composite Curve Showing the Moisture Content of Five Woods at Different Humidities and Ordinary Room Temperatures
(From "Wood in Aircraft Construction," Forest Products Laboratory)

Gluing propellers. After the kiln drying process is completed and the lumber is brought down to a moisture content between 5 and 7 per cent, the boards are planed down with extreme accuracy to a uniform thickness before gluing. Pat-

[1] For a detailed reference on the subject of Kiln Drying, see "Wood in Aircraft Construction," Forest Products Laboratory.

terns corresponding to the shape of the various laminations are then marked out on the boards and they are sawed to form the shape of the patterns. The boards are then glued together in proper relation and held in place by means of a *C* clamp or other special clamping device until the glue has set. This usually requires about twelve hours.

For gluing propellers, animal hide glue is most commonly used, although experiments have been made with casein and blood albumin glue.[2]

Shaping of propellers. The process of shaping a propeller is carried out almost entirely by hand where only two or three propellers of one design are to be made. For production purposes where a design is to be duplicated in considerable quantity, a number of different types of duplicating machines are available which will work the propeller down within about one-eighth of an inch of the finished size. After the rough shaping, the propeller is attached to a board cut to the exact contour of the projected planform of the blade; then the propeller with its pattern is run around a wood shaper cutting out the contour of the blades.

The remainder of the process of shaping is carried out by hand with spoke-shaves and hand scrapers, the propeller being finally rubbed smooth with sandpaper. The flat face of the blade is worked down very accurately and checked by means of protractors as the work proceeds, the back face being cut down to fit templates at various stations.

Finish of propellers. In finishing, the propeller is first coated with a suitable filler so that the varnish will not sink into the pores of the wood; it is then given one coat of spar varnish.

After the varnish becomes tacky the propeller is coated all over with aluminum leaf which is only about one, one hundred thousandth (0.00001) part of an inch in thickness. In spite

[2] As a reference on gluing, see "Conditions Affecting Making of Glued Joints," by T. R. Truax in the *Furniture Manufacturer and Artisan*, May, 1924.

of the extreme thinness of this material, it provides a very excellent seal against moisture penetration so that the changes in moisture content are very very slow, resulting in no damage to the material.

After the application of the aluminum leaf, the propeller is given two coats of maroon enamel followed by a final coat of varnish to give a smooth finish.

The shade of maroon enamel adopted by the Army Air Service is the result of extensive experimental development in

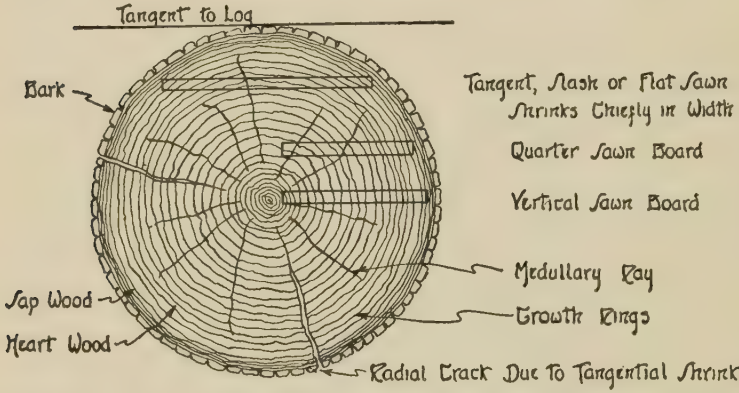


Figure 92. Illustration of Various Methods of Sawing Lumber from the Log

order to provide a finish giving the least possible glare from the flood lights thrown on it while making landings at night. A propeller coated with grey or white enamel under these conditions would create an opaque disc before the pilot so that he could not see anything ahead in landing. The maroon enamel, on the other hand, gives fairly satisfactory vision and for this reason has been standardized for all Air Service propellers.

Kinds of woods used. A number of woods have been used in propeller construction; birch, walnut, Honduras mahogany, white oak, and cherry being among the preferred woods.

All propeller lumber should preferably be quarter-sawed as

the quarter sawed material has less tendency to warp than the slash-sawed.

For very high-speed propellers, birch is the preferred wood owing to its great strength and life under the whipping action to which the propeller is subjected. For moderate speeds, oak, walnut, and cherry are all suitable. Walnut is particularly good on account of its combining a comparatively great strength with relative freedom from warping.

Quartered white oak is somewhat more inclined to warp, but it makes about as strong a propeller as walnut and seems to have the advantage of standing water spray with less abrasion than any of the other woods. For this reason white oak has found extensive use in the Naval Service for seaplanes.

Mahogany is only suitable for very low-speed propellers as it is a rather weak wood, but it is considerably lighter than any of the other woods mentioned. It is very free from warping tendencies, and on account of light weight it is suitable for very large slow-turning propellers.

In cases where the size of the propeller is extremely large, even mahogany propellers become too heavy. The writer has devised a method of building a very large propeller with a core of balsa wood covered with a number of layers of mahogany veneer. This construction gives a very light propeller and apparently a very strong one.

Micarta Propellers

Characteristics. Micarta is a trade-name adopted by the Westinghouse Electric & Mfg. Co., for a laminated product made up of plies of fabric or paper previously treated with a phenolic condensation product known as *bakelite,* and subjected to heat and pressure to form a solid mass.

The material has a specific gravity of about 1.37, a tensile strength of about 9,000 pounds, per square inch, and compressive strength of about 20,000 pounds, per square inch. The

modulus of elasticity is very low, running between 1,000,000 to 2,000,000. The strength of the material can be controlled to a large extent by the weave of the fabric, the percentage of bakelite and pressure applied in molding.

While the tensile strength is rather low compared to the weight of material, it is probable that this value can be doubled by the use of specially woven fabric.

Micarta appears to have the property of rapidly damping harmonic vibrations and for this reason makes a very strong propeller in spite of the comparatively low mechanical strength properties.

It is entirely free from corrosion or any weathering effects and the material in this respect is superior to either wood or metal.

Several designs of micarta propellers have been developed by the writer and built by the Westinghouse Electric & Mfg. Co., of East Pittsburgh, Pa.

The first designs were made in one piece on account of the relative simplicity of this type of construction.

The mold required for building these propellers is extremely expensive however, and later designs have been made with detachable blades which can be adjusted for any pitch angle so that a considerable range of engine power and airplane speed can be covered with one design.

In manufacturing micarta propellers the methods differ considerably from those employed with wood propellers on account of the hardness of the material. The propellers must be formed accurately to size in a mold and afterwards balanced without the customary trimming of the material from the tips.

Press and mold for micarta propellers. In constructing the mold, because of the peculiar shape of an air propeller, considerable handwork is necessary. For this reason a metal was selected which can be readily chipped, filed, scraped and polished, and at the same time have sufficient hardness to give

the mold satisfactory life. A high-grade phosphor bronze was tried and has proved very satisfactory. The high heat conductivity of this material gives the mold quick heating and cooling properties.

The mold is heated with steam and cooled with water. It is of sectional construction to permit insertion of the brass heating pipes and also to facilitate machining.

After removal from the mold the propeller requires the following operations:

(1) Filing edges
(2) First boring the propeller boss
(3) Preliminary balancing
(4) Final boring of boss
(5) Cutting keyways
(6) Installing hub
(7) Curing
(8) Final balancing
(9) Finishing

Mechanical properties. The following tables enumerate the strength characteristics of micarta.

MECHANICAL PROPERTIES OF VARIOUS KINDS OF MICARTA

	Tensile	Compression Edgewise	Transverse Flatwise
Micarta duck	9,560	21,000	20,500
Micarta-draft paper	15,600	22,000	25,000
Micarta-special fabric	26,000 [a]	22,800 [b]	39,300 [a]
Micarta-wood veneer	36,000 [c]	7,970 [d]	41,300 [c]

[a] Warp threads lengthwise of test specimen
[b] Parallel to warp thread
[c] Grain lengthwise of test specimen
[d] Parallel to grain

Material	Ultimate Tensile Strength	Modulus 0 to 5,000 lbs./sq. in.
Micarta-wood	36,000	3,200,000
Micarta paper	15,600	3,270,000
Micarta special fabric	26,000	2,000,000
Micarta cotton duck	9,560	1,180,000

Micarta threads. A number of tests have been made to determine the shearing strength of micarta threads. This data

SHEARING STRENGTH OF MICARTA THREADS [3]

No.	Micarta	Threads per in.	Length	Diameter outside	Root diameter	Thread Area at root	Yield lb.	Yield sq. in.	Maximum load	Ult. Str. lb. per sq. in.
2	Wood	6	2	3	2.74	15.06	46,000	3,054	85,000	5,644
4	Canvas	8	2	3	2.80	15.39	57,000	3,704	92,000	5,978
5	Canvas	4	2	3	2.62	14.40	50,000	3,472	78,700	5,465
10	Canvas	2	2	4	3.25	17.87	55,000	3,078	100,000	5,596 *
11	Canvas	6	2	4	3.74	20.56	58,500	2,845	100,000	4,864 †
12	Canvas	4	2	4	3.62	19.90	37,000	1,859	100,000	5,025 ‡

* 100,000 pounds for 28 hours
† 100,000 pounds for 15 minutes
‡ 100,000 pounds for 4 minutes

[3] Abstracted from N. A. C. A. Technical Note No. 200.

is useful for the purpose of design. The thread used in each case was the trapezoidal type; the diameter of the piece and the number of threads per inch being shown in the following table. The shearing strength of the material in these threads ran about 5,000 pounds per square inch in each case.

Micarta propellers have been used to a considerable extent by the Army and Navy Air Service and have been particularly successful in Panama where the humid climate is especially severe on wooden propellers.[4]

Metal Propellers

The experimentation with metal propellers has been carried out along three more or less distinct lines.

1. The tubular type made from a tube pressed into shape to give the desired cross-section for the blades.
2. The built-up or fabricated type in which a number of sheets or structural members are combined to form the propeller.
3. The solid type in which the blades are made of solid metal.

Tubular type of metal propellers. Just at the present time there is not very much activity with the tubular type of metal propeller, although a number have been built and some of them have been fairly successful. The manufacturing difficulty in this type of propeller has been rather severe, and it is largely for this reason that the development is not very active at the present time. Among the earliest metal tubular propellers tested in the United States was the J. W. Smith steel propeller.

The most successful of these consisted of a steel tube turned with a tapering wall from the hub to tip. The only propellers built were for the Liberty engine, having a wall of about five-sixteenths of an inch at the hub and tapering down to a wall of about one-sixteenth of an inch at the tip. The two blades of

[4] As a reference on Micarta Propellers, see N. A. C. A. Technical Note No. 200, "Micarta Propellers," by F. W. Caldwell and N. S. Clay.

this propeller were made separately and screwed into a metal hub, being held in place by the trapezoidal or buttress type of thread. For this reason the blades were left round at the hub and were pressed out to shape at the outer portions. At the outer end of the blade the leading edge of the tubular section was cut away in order to give a tapering tip. This leading edge was then welded. These propellers were provided with internal lattice work bracing extending along the length of the blade and fastening to the tube by means of rivets. They proved to be fairly strong, but were rather heavy and not very efficient owing to the difficulty of providing an accurate cross-section shape.

The second type of tubular construction was the Dicks propeller, manufactured by the Standard Steel Propeller Company. In the case of this propeller a single tube was used, tapered from the hub to each end. These tubes were pressed in an accurately constructed die; the tubes being forced to conform to the shape of the die by means of oil pressure inside of the tube. This construction gave a fairly accurate shape to the blades. The only welding was across the end of the tip where it was necessary to close the tube. The resulting propeller was quite light, but did not have a sufficient factor of safety owing to the thin wall of the tube. It was not feasible to use a wall much thicker than one-eighth of an inch on account of the difficulty in forming and the tremendous oil pressure necessary to break down the material around the hub portions. This process proved extremely expensive, particularly in the experimental stages.

The Standard Steel Propeller Company later introduced a tubular type of reversible propeller for the Liberty engine. In this case both the diameter and the wall were tapered so that the only welding required was across the end of the narrow tip. For these propellers the internal portion had to be reamed out with a special reamer, and the outside turned down to form. After the tubes were turned up, the blades were placed in a

special press. A die being employed to form the external contour and a special form of mandrel being used to hold the internal shape. By this process a reversible propeller was produced which appeared to have adequate strength. However, the propeller produced in this way was rather heavy, extremely expensive, and never very extensively used.

Fabricated metal propellers. A great many types of fabricated metal propellers have been constructed, but the only one which has come into any general use is the Leitner Watts type

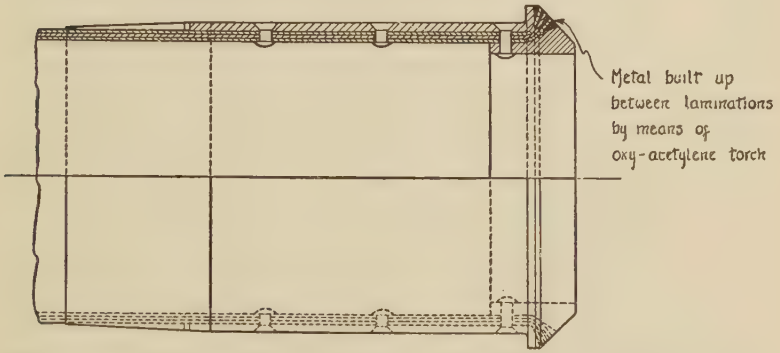


Figure 93. Method of Forming Retaining Shoulder on Cylindrical End of Leitner Watts Propeller

developed in England. This propeller is made up of a number of thin sheets one above the other, one set of sheets being used to form the back face of the blade and the other to form the pressure face. The sheets for these two faces were pressed out in dies and were welded together along the leading and trailing edges. In the latest construction the blades are made up separately from the hub, having a cylindrical shape at the hub end so that they may be set at any pitch angle when assembled in the hub. The centrifugal force is taken up by a single shoulder at the end of the blade, a very ingenious method being used to form this shoulder. The individual sheets are bent out somewhat as shown in Figure 93, and material is built up between

the sheets by means of the oxy-acetylene torch. The hub proper consists of a central plug made up externally to a standard form and bored internally to fit the particular engine on which it is to be used, and a separate cylindrical barrel in which the two blades are mounted. The two halves of this barrel are held together by bolts clamping the blades in position so as to avoid any tendency to work loose.

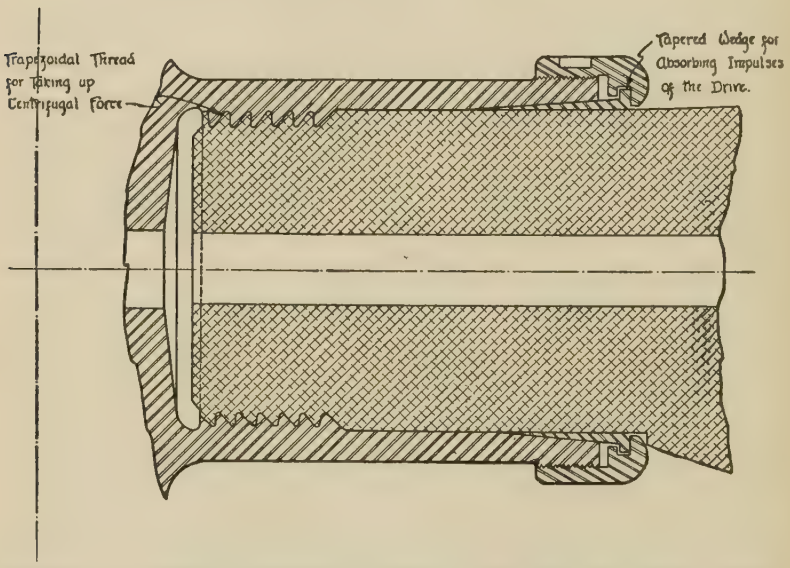


Figure 94. Method of Attaching Aluminum-Alloy Propeller Blade to the Steel Hubs as Used by the Standard Steel Propeller Company

The Leitner Watts propeller appears to be quite strong, having been subjected to severe overload tests. The aerodynamic efficiency is about equal to that of the wooden propeller. This type has come into quite general use in England and France, but has not been adopted very extensively in the United States on account of the development of the more efficient solid metal types.

Solid metal propellers. The development of a solid metal propeller was begun by the writer in the laboratory of the

Engineering Division during the early part of 1918. The propellers of the early experiments were made of steel and seemed to show considerable promise from the start. The perfection of the strong and light aluminum alloy however, has largely superseded the use of the steel in the solid propellers, except for extremely high engine speeds.

The Standard Steel Propeller Company has continued the early experiments made on solid metal propellers at the Engi-

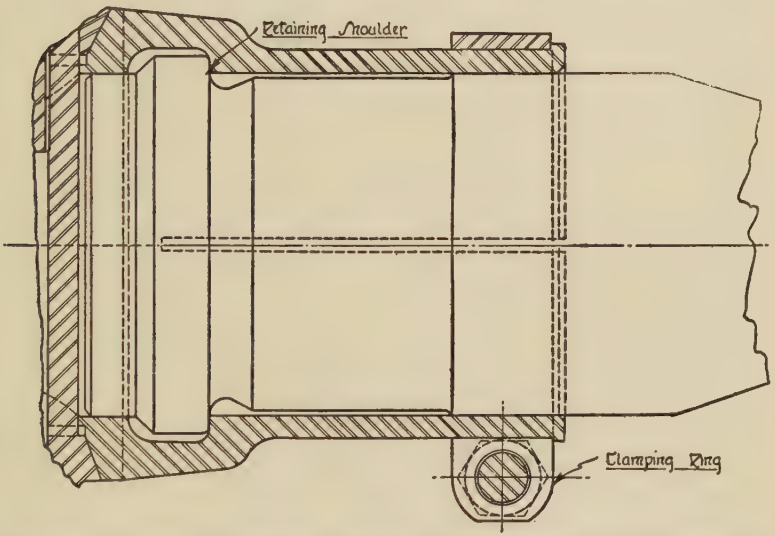


Figure 95. Method of Attaching Propeller Blades into Split Hub Showing the Retaining Shoulder and Clamping Ring

neering Division, and has developed a series of aluminum alloy propellers in which the blades are drop forged to substantially the finished shape; they are held into the hub by means of a novel wedge device. The centrifugal force is taken up by a trapezoidal thread at the end of the blade; the drive is carried out by means of a cylindrical wedge which is forced in place between the hub and the blade, drawing it tight by a collar which is threaded on to the outside of the hub. These solid metal propellers show an efficiency very much higher than the

usual wooden types, often giving an increase in speed of about 5 per cent as compared to the wooden propellers. Owing to the pitch adjustment feature they are quite adaptable—one propeller can be used on a large number of planes.

Considerable care must be used in the manufacture of these propellers, from the casting of the billets, through the forging process down to the finished article. As it is particularly desirable to avoid forging seams, the billets are ground all over before being put under the hammer. The hub end of the blade is rough formed in a special die and the outer portion drawn out to rough proportions under an ordinary drop hammer. The rough forging is then placed in the die and brought to shape by hammering down, the flash being removed by means of a trimming die. Under the hammer most of the material flows out through the tip giving a very desirable structure. Contrary to the general impression these aluminum-alloy forgings require about as large a hammer as the steel forgings of the same size since the aluminum alloy is forged at a very much lower temperature. The alloy used in these propellers at the present time is designated by the trade-name of "25-S" by the Aluminum Company. This alloy may be quenched from a suitable temperature after forging and remains soft indefinitely until subjected to an artificial aging process. All the work on the blades may be done in a soft condition as the aging process is carried out at a very low temperature and does not distort the material. The aging is carried out for approximately 48 hours at a temperature about 350° Fahrenheit.

On account of their adaptability, these detachable blade aluminum alloy propellers are finding a considerable application in the Army and Navy Air Services. They are quite expensive in small quantities, but doubtless will be very cheap in extremely large quantities.

A second type of solid metal propeller is the one developed by S. Albert Reed. In this construction a single sheet of

aluminum alloy is given a double twist so that the flat face of the sheet assumes the correct helical angles. These propellers are machined down to contour either before or after twisting. The present practice is to machine out the leading and trailing portions of the propeller blades by means of a milling cutter forming an approximation to an airfoil section by this method.

PART III

WATER BALLAST RECOVERY

By

ROBERT F. KOHR, M.E.

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CHAPTER XII

DESIGN CONSIDERATIONS

Practicability

Purpose of water ballast recovery. The only practicable means thus far developed for avoiding the valving of lifting gas, to compensate for the increased lift due to the consumption of fuel in airship engines, is the condensation of a large part of the water of combustion formed by the combination of the hydrogen in the gasoline with the oxygen of the air. When ordinary aviation gasoline is used, about 140 pounds of water is formed in burning 100 pounds of fuel, consequently the condensing and separating apparatus need not be designed to extract all of the moisture from the exhaust gases.

Experiments conducted at the Bureau of Standards for the United States Army Air Service, as well as later flying tests of condensing apparatus on the U. S. Navy Airship "Shenandoah," have demonstrated the feasibility of replacing the fuel burned by an even larger weight of ballast water, by the relatively simple expedient of cooling the engine exhaust gas to within a few degrees of air temperature and separating the entrained moisture from the gases as they leave the apparatus.

Weight and head resistance. The weight of water recovery apparatus for airships intended for long-distance flying need not give the designer any concern. It has been demonstrated that the weight of the essential equipment can be kept within limits of 1.5 pounds per brake horsepower of the engines served; this weight can be wholly compensated by reduction of the ballast normally carried. The initial ballast replaced by the condensing apparatus will reproduce this weight of condensed water in the time necessary to burn an equivalent weight of

fuel, or in about 2½ hours if the engines are operated at full power. Flying tests on the Army Airship D-3 have shown no appreciable increase in head resistance due to the installation of apparatus of the type herein discussed.

Theory

Nomenclature. In reviewing the theory involved in the process of obtaining water from the engine exhaust gas, it may

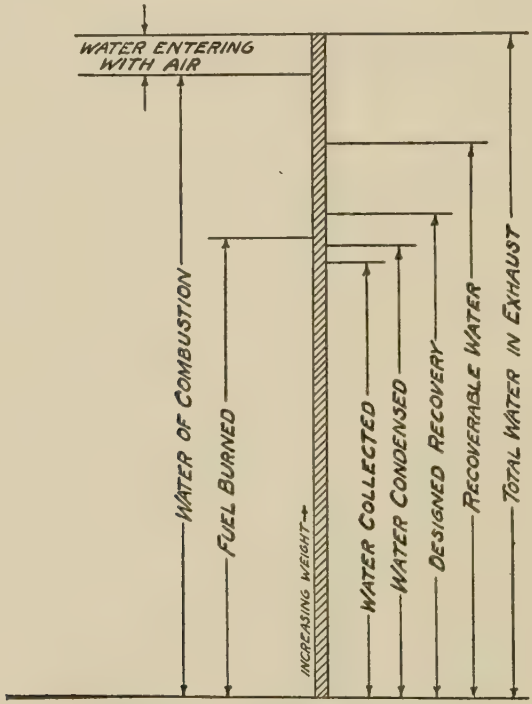

Figure 96. Nomenclature Chart

be well, in order to avoid confusion, first to discuss the nomenclature used.

Reference is made to the chart, Figure 96, in which vertical distances represent weights of the quantities indicated.

The fuel burned is, of course, the basis upon which the other quantities are determined. The total water in the exhaust gas is the sum of the water of combustion and the moisture entering with the carburetor air. This quantity depends upon the kind of fuel and, to a small extent, upon the mixture ratio (air to fuel) and relative humidity.

Since the amount of water lost as vapor depends upon the final temperature of the exhaust gas, and since in an air-cooled system, the limit of cooling is the air temperature itself, at once it becomes apparent that the recoverable water is the total minus the vapor necessary to saturate the gaseous exhaust products at air temperature.

The designed recovery is, of course, limited by the recoverable water as determined by an assumed air temperature. It may be arbitrarily placed at any value less than that of the recoverable water, but would not ordinarily be less than the fuel burned.

Since the estimated recovery is based upon an assumed final temperature, it is evident that error in design assumptions or changes in flying speed or air temperature will change this final temperature and therefore the amount of water actually condensed. The water condensed may be either greater or less than the designed recovery, but is represented in Figure 96 as showing a loss.

In the collection of the water condensed, leaks of any kind, as well as faulty separation of entrained moisture, will lower the quantity available as ballast. The water collected is therefore always less than that actually condensed.

The quantities shown, when expressed as per cent, are based upon the weight of fuel burned.

Design. The design of the condensing apparatus is dependent upon a large number of variables, which may be taken up best as they appear.

It is apparent at once that the water condensed is dependent

upon the composition of the exhaust gas and its final temperature. The first of these is readily determined when the chemical composition of the fuel, ratio of air to fuel, and relative humidity and temperature of the air, are known. However, the final exhaust temperature is a function of the length, diameter, spacing and number of condenser tubes, mass flow and initial temperature of exhaust gas, and the velocity and temperature of the air flowing past the tubes.

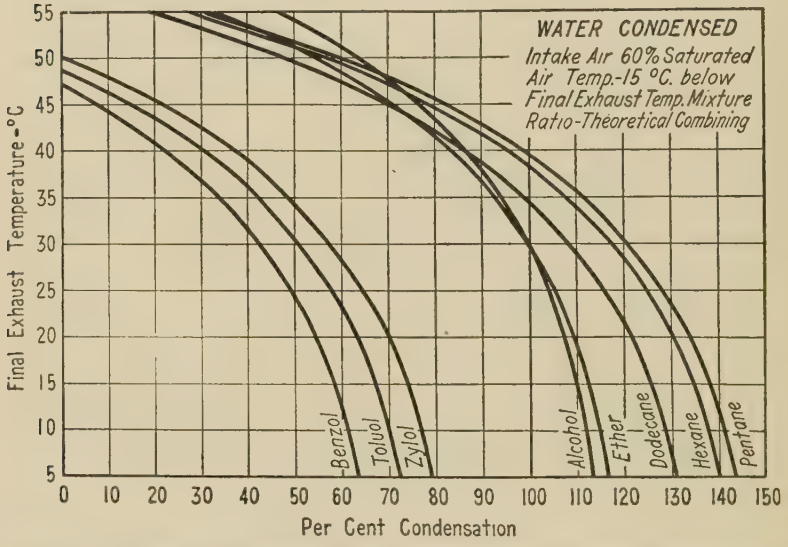

Figure 97. Water Condensation from Various Fuels of Average Composition

Influence of fuels. The influence of different fuels is shown in Figure 97, in which the water condensed is plotted against final exhaust temperature for each of the fuels considered. These curves were obtained by adding the water of combustion to the water entering with the carburetor air, and subtracting the weight of vapor necessary to saturate the gaseous exhaust products at the temperatures indicated. The entering air temperature in each case was assumed to be 15°C. lower than that of the cooled exhaust gas, and its relative humidity

was taken as 60 per cent, which is an approximate average for all parts of the United States for the whole year, for altitudes less than 5,000 feet. As a basis of comparison of the different fuels, their chemically correct air-fuel ratios were used.

Since it is improbable that any fuel having exactly the composition of one of those shown in Figure 97 will ever be put into service use, this figure is supplemented by Figure 98, show-

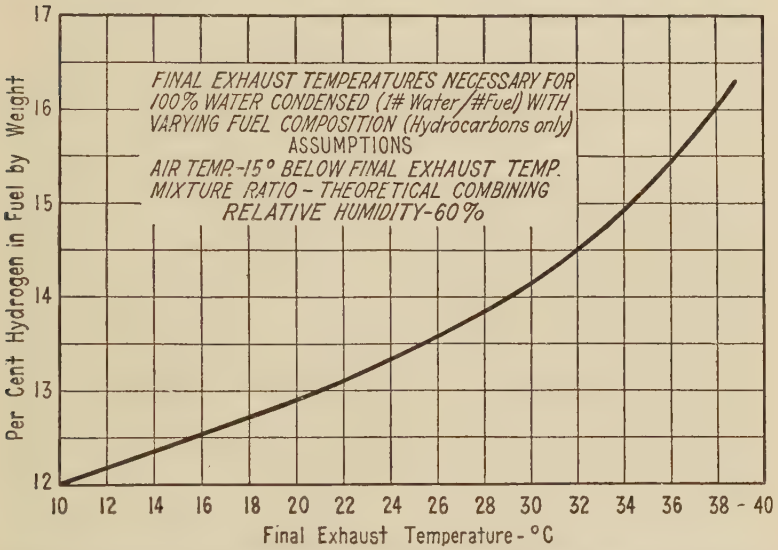
FINAL EXHAUST TEMPERATURES NECESSARY FOR 100% WATER CONDENSED (1# Water/#Fuel) WITH VARYING FUEL COMPOSITION (Hydrocarbons only)
ASSUMPTIONS
AIR TEMP.-15° BELOW FINAL EXHAUST TEMP.
MIXTURE RATIO - THEORETICAL COMBINING
RELATIVE HUMIDITY-60%
Per Cent Hydrogen in Fuel by Weight
17
16
15
14
13
12
10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 - 40
Final Exhaust Temperature - °C

Figure 98. Final Temperature to which Exhaust Gas must be Cooled to Condense 1 pound of Water per pound of Hydrocarbon Fuel Burned

ing the final temperature to which the exhaust gas must be cooled in order to condense 1 pound of water per pound of hydrocarbon fuel burned. This curve was derived by plotting the points where curves such as those of Figure 97 cross the 100 per cent condensation line.

Mixture ratio. All of the curves thus far shown are based on theoretical combining mixture ratios of air to fuel, which correspond to extremely lean carburetor settings and give the economy so vitally important in the operation of airships.

Some multi-cylinder engines are unable to operate on mixtures quite as lean as those shown, but since the leaner mixtures are less advantageous as regards water recovery, the curves are based on the worst expected condition, and should be considered as on the safe side.

The effect of mixture ratio upon water condensed is shown by Figure 99. The fuel in this case was assumed to be gaso-

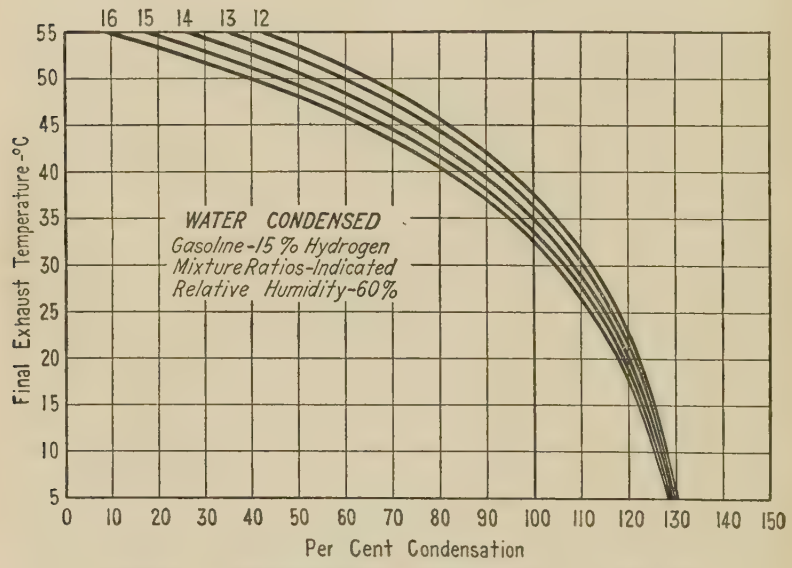

Figure 99. Effect of Mixture Ratios on Water Condensed

line having a 15 per cent hydrogen content, which closely approximates the fuels now in service use. Figure 100 shows the effect of mixture ratio upon the final temperature to which the exhaust gas must be cooled in order to condense 1 pound of water per pound of gasoline burned.

Humidity. Since the humidity is uncontrolled it is possible that for special cases some value other than the average one of 60 per cent should be applied. The curves of Figure 101 indicate the limits for the humidity range between 0 and 100 per

cent. It will be seen that at a temperature corresponding to 100 per cent condensation with saturated air, the decrease caused by a change to dry air is 29 per cent. The points plotted in this figure are average values obtained during a 90-hour endurance test of the Model I (Bureau of Standards) condenser. These points represent actual water collected, and

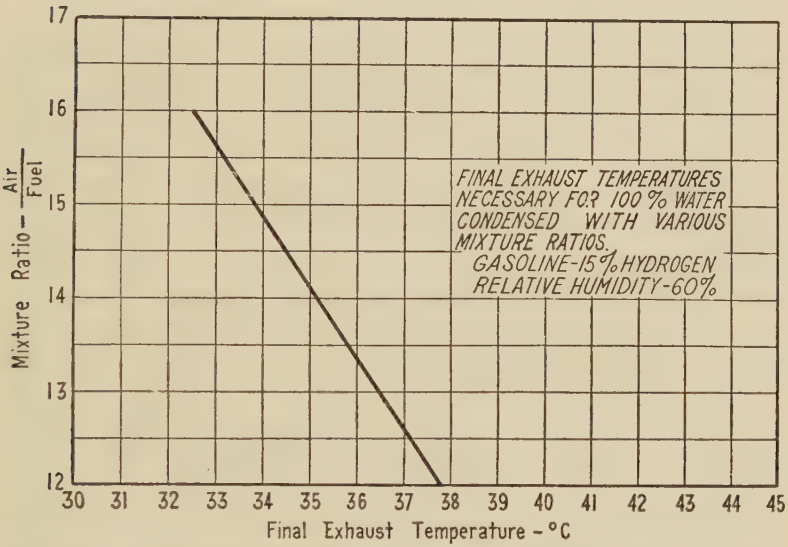
Mixture Ratio - Air/Fuel
17
16
15
14
13
12
30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45
Final Exhaust Temperature - °C
FINAL EXHAUST TEMPERATURES NECESSARY FOR 100 % WATER CONDENSED WITH VARIOUS MIXTURE RATIOS.
GASOLINE-15% HYDROGEN
RELATIVE HUMIDITY-60%

Figure 100. Effect of Mixture Ratio upon Final Temperature to which Exhaust Gas must be Cooled to Condense 1 pound of Water per pound of Gasoline Burned

it will be readily recognized that faulty separation of entrained moisture or any leakage of either exhaust gas or water would lower the apparent condensation (water collected).

Heat transfer. It is obvious from the foregoing paragraphs and curves, that it is a relatively simple matter to determine a desirable final exhaust temperature, having assumed the kind of fuel, mixture ratio, air temperature and relative humidity. From this point on, the task is to so select and

arrange the cooling system as to arrive at the desired final temperature.

The equations for heat transfer which are the basis of the design of the exhaust cooling system have been derived under the following assumptions: (1) the surface temperature of the pipe is the same inside and out at any given point in the length of the pipe and is uniform around the circumference; (2) the

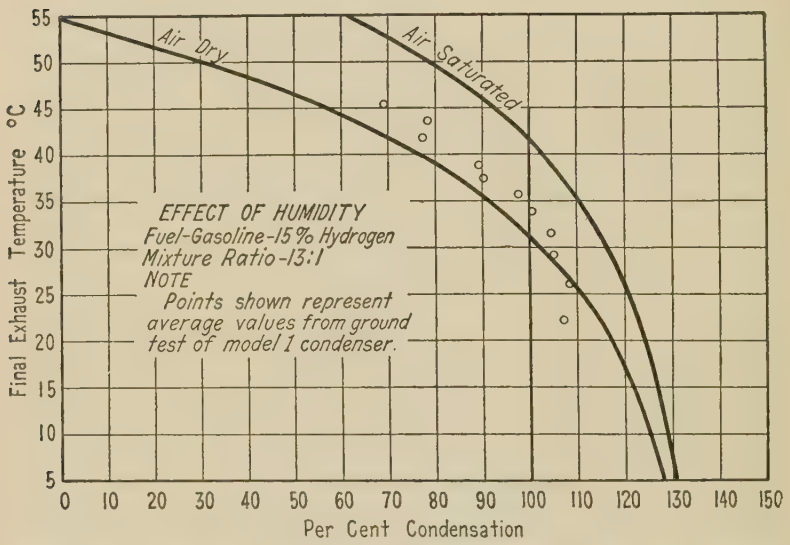

Figure 101. Limits for the Humidity Range between 0 and 100 per cent

cooling air flows with uniform velocity in a *direction parallel to the axis of the pipe,* is turbulent, and of temperature, density, pressure and humidity characterizing normal atmospheric conditions.

The resulting equations arranged in a form convenient for design computations are:

For counterflow:

$$\log_e \frac{T_a - t_b}{T_b - t_a} = p \frac{q_h q_c}{q_h + q_c} \left(\frac{1}{H_h} - \frac{1}{H_c} \right) L \qquad (1)$$

For parallel flow:

$$\log_e \frac{T_a - t_a}{T_b - t_b} = p \frac{q_h q_c}{q_h + q_c} \left(\frac{1}{H_h} + \frac{1}{H_c} \right) L \qquad (2)$$

in which

T_a = Initial exhaust temperature °C.
T_b = Final exhaust temperature °C.
t_a = Initial air temperature °C.
t_b = Final air temperature °C.
p = Perimeter of tube, centimeters
L = Length of tube, centimeters

The terms H_h and H_c are obtained from the relations:

$$H_h = M_h C_h$$
$$H_c = M_c C_c$$

in which

H_h = heat capacity of the mass of exhaust gas which passes any point in unit time in $\frac{\text{calories}}{\text{second} \times \text{°C.}}$

M_h = mass of exhaust gas passing any point along the pipe in unit time, grams per second

C_h = specific heat of exhaust gas at constant pressure, $\frac{\text{calories}}{\text{gram} \times \text{°C.}}$

H_c = heat capacity of the mass of air stream which passes any point in unit time $\frac{\text{calories}}{\text{second} \times \text{°C.}}$

M_c = mass flow of air past any point along the pipe, grams per second

C_c = specific heat of air at constant pressure, $\frac{\text{calories}}{\text{gram} \times \text{°C.}}$

The terms q_h and q_c are obtained from the Lanchester-Stanton relation[1] combined with Lees' equation for skin friction

$$q = C_p m \left[0.0765 \left(\frac{\mu}{MD} \right)^{0.35} + 0.0009 \right] \qquad (3)$$

[1] Great Britain Committee for Aeronautics, Tech. Report 1912–13, page 45; Bureau of Standards Technologic Paper 211, page 320.

in which

C_p = Specific heat at constant pressure
μ = Viscosity, poises
M = Mass flow, $\frac{\text{grams}}{\text{second} \times \text{sq. cm.}}$
D = Diameter of tube, centimeters

In applying these equations it is necessary to determine a value for the equivalent diameter of tube through which the air may be considered to flow. In a condenser consisting of a bank of tubes, parallel to the air stream, it is evident that the spacing of the tubes determines the dimensions of the space surrounding any tube through which the corresponding air stream flows. Hence, if the vertical and horizontal spacing be taken as A and B respectively the cross-sectional area of air passage will be $AB - \frac{\pi d^2}{4}$ where d is the diameter of the tube. The resulting figure for the area will then apply in determining the mass flow for the air surrounding one tube. To determine the equivalent diameter for use in determining q_a in equation (3), an approximation can be made readily by assuming that the conditions of flow at the cooling surface are the same as might be expected if the air were actually flowing through a tube of the same cross-sectional area as the space actually occupied by the air stream. The equivalent diameter D, will then be

$$D = \sqrt{\frac{4A}{\pi}} \text{ where } A = \text{area.}$$

Design Considerations

In making the computations for a satisfactory heat dissipating apparatus the first determination necessary is the total rate of mass flow of exhaust gas coming from the engine. This will be the product of the brake horsepower, the specific fuel consumption (pounds per brake horsepower hour), and

the weight of the mixture taken by the engine per pound of fuel, giving the mass flow in pounds per hour.

For maximum efficiency of heat transfer, it is desirable that the velocity of the exhaust gas and the cooling air, relative to the pipe wall, be of the same order of magnitude. As a first approximation, the velocity of the exhaust gas may be taken as equal to the flying speed. However, the density of the exhaust gas changes by a factor of 3 as the temperature drops from 650° to 30°C., so that for a given mass flow the velocity at the hot end is three times that at the cold end. Thus, for an air speed of 45 miles per hour, the average cruising speed of the Army airships now in service, the corresponding velocity at the hot end would be 135 miles per hour, which threatens excessive exhaust back pressure. A compromise figure for mass flow inside the pipes of 0.017 pounds per second per square inch of cross-sectional area (1.195 grams per second per square centimeter) has been found satisfactory, both as to heat transfer and back pressure.

The selection of the most advantageous size of tubes must of necessity be a compromise between heat transfer efficiency and mechanical considerations. The effectiveness of the apparatus will be found to be increased as the diameter of the tubes is decreased, whereas the smaller the tubes, the greater will be the exhaust back pressure and liability to clogging, and the less the mechanical strength of the tubes, both as self supporting beams and as long columns.

An indication of the relative resistance to gas flow with increase or decrease of tube size may easily be obtained, but in computations of this kind it is safe to let the limiting factor in reducing tube diameter be the mechanical strength necessary in a structure of the type contemplated. In condenser banks not more than 10 feet in length, aluminum tubing of 1 inch diameter with the thinnest walls commercially obtainable has been found satisfactory both as to mechanical strength and back pressure.

Having determined the tube size and desirable mass flow per unit area, the total number of tubes and mass flow per tube are easily computed.

Inasmuch as the Lanchester-Stanton equation for film coefficients is applicable strictly only to smooth round pipes, it is evident that it cannot be expected to give exact results when used to determine the heat transfer from the outside of the tubes to the air stream. However, in the absence of a more reliable method the equation has been used in this way. The results seem to justify such use.

Specific heat. Since the exhaust gas consists of approximately 10 per cent steam, for which $C_p = 0.45$, and 90 per cent other products of combustion for which $C_p = 0.24$, the equivalent C_h may be taken as 0.26.

For the range below 53°C. the effective specific heat is very high, owing to the latent heat of condensation. This has been determined approximately by computing the total heat of 1 gram of the gas mixture at frequent temperature intervals below 53°C. and finding the rate of change of total heat with temperature. This rate of change may be taken as the effective specific heat with a mean value of 2.83 in the range 53°C. to 45°C. and 1.92 in the range 45°C. to 35°C., or 2.20 for the entire range 53°C. to 35°C. The figure 45°C. is an approximate one for the temperature at the end of the second bank in a three-bank condenser, of which the first and last banks are counterflow and the center one parallel flow. This temperature is used only for determining the effective specific heats in the ranges indicated.

Viscosity. The viscosity of the gas changes considerably with temperature, but an average value of μ over the range 600° to 53°C. may be taken as 2.4×10^{-4} poises, equal to the viscosity of air at 150°C. For the range below 53°C. it may be assumed that the effect of the condensing vapor is negligible and the value 2.0×10^{-4} poises may be used. For the air

flowing outside of the tubes, $\mu = 1.85 \times 10^{-4}$. Having these constant values for the variables in equation (3), q_h and q_c and consequently the function $\frac{q_h\, q_c}{q_h + q_c}$ may be taken as constant for each range above and below the condensation point.

Computation method. For a condenser of more than one bank it is impossible to compute the length directly, on account of the changing functions of the variables during parallel and counterflow as well as the change of C_p with temperature of the exhaust gas. The most convenient method is to assume the length, then solve equations (1) or (2) for each section. Of course it is necessary to measure or assume the temperature of the exhaust gas received by the condenser as well as the air temperature.

It will be found that an equation in two variables will be left in the case of each section considered, but that one of these can be eliminated by solving simultaneously with the equation expressing the fact that the heat lost by the exhaust is equal to the heat taken up by the air

$$(T_a - T_b)\, H_h = (t_b - t_a)\, H_c$$

The curves of Figure 102 show air temperature plotted against necessary bank length of a three-bank condenser, for three air speeds and four initial exhaust temperatures. These summarize the effect of changes in the most important variables on heat transfer, for a constant final exhaust temperature of 32°C. This figure represents the conditions necessary for the condensation of 1 pound of water per pound of gasoline burned. The number of 1 inch parallel tubes is taken as one for each six horsepower of the engine or engines served.

Prospective improvement. Research has been conducted with a view to using the fabric cover of a rigid airship for the final stage of cooling. Inasmuch as the actual condensation takes place entirely below 60°C. the temperature is too low to result in damage to the fabric; furthermore, tests of long

duration have shown that cool exhaust gas has no appreciable effect upon the durability of fabric. This seems reasonable, in view of the fact that exhaust gas usually contains no oxygen.

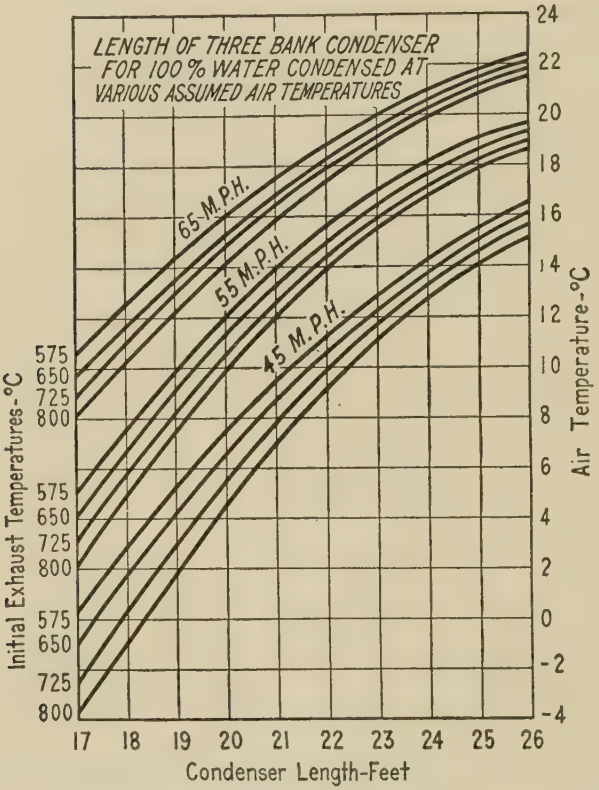

Figure 102. Relations of Air Temperature to Bank Length for a Three-Bank Condenser, at Three Air Speeds and Four Initial Exhaust Temperatures

Experiments on an airship in service have demonstrated that the use of fabric for the latter stages of cooling is feasible and it results in a saving of weight.

CHAPTER XIII

SOLUTION OF A DESIGN PROBLEM

A Typical Design Problem

For purposes of illustration, the design of a condenser for a class "D" airship is considered, assuming:

Maximum B.hp	= 300
Fuel	aviation gasoline
Cruising speed	45 m.p.h.
Mixture ratio	15 : 1

Referring to Figure 99, the 15 : 1 mixture ratio curve crosses the 100 per cent condensation line at a temperature of 33.5°C. This is the final temperature to which the exhaust gas must be cooled.

Mass flows. The total mass flow of exhaust gas, based on an assumed specific fuel consumption of 0.50 pounds per B.hp. hr., is

$$0.50 \times (15.0 + 1.0) \times 300 = 2{,}400 \text{ pounds per hour}$$

For a speed of 45 miles per hour, a mass flow of 0.017 pounds per second per square inch cross-sectional area has been found satisfactory. Assuming round tubes of 1 inch diameter, the cross-sectional area of each is $\frac{\pi}{4}$ square inch.

The mass flow per tube is then

$$0.017 \times \frac{\pi}{4} = 0.0133 \text{ pounds per second}$$

Hence, the number of tubes necessary is

$$\frac{0.667}{0.0133} = 50$$

The rise in temperature of the cooling air must be kept small

if the final temperature of the cooled exhaust is to be low. The mass flow of air, therefore, must be much greater than that of the exhaust gas, and since the mass flow of air per tube is controlled by the tube spacing this must be assumed in such a way as to secure a compromise between extremely high mass flows and extreme compactness of the apparatus.

In the design of one of the first exhaust water condensers built, the tube spacing, both horizontally and vertically, was 2¼ inches. Assuming this spacing, the cross-sectional area of the air space is

$$2\tfrac{1}{4} \times 2\tfrac{1}{4} - \frac{\pi}{4} = 4.277 \text{ square inches per tube}$$

The mass flow of air is

$$M_c = \rho V A$$

in which

ρ = Density = 0.0754 pound per cubic foot (750 mm. Hg., 15°C.)

V = Velocity = 45 m.p.h. = 66 feet per second

A = Cross-sectional area = 4.277 square inches = 0.0297 square foot

$M_c = 0.0754 \times 66 \times 0.0297 = 0.148$ pound per second = 67 grams per second

Heat capacities. From page 187 the specific heats are

$C_c = 0.24$

$C_h = 0.26$ above 53°C.
$= 2.83$ between 53°C. and 45°C.
$= 1.92$ between 45°C. and 35°C.

Hence, numerical values for heat capacities are

$$H_c = M_c C_c = 67 \times 0.24 = 16.08$$

and

$H_h = 6 \times 0.26 = 1.56$ above 53°C.
$= 6 \times 2.83 = 17.0$ between 53°C. and 45°C.
$= 6 \times 1.92 = 11.5$ between 45°C. and 35°C.

In this three-bank type of condenser the initial condensation point, 53°C., will come somewhere in the second bank, and the remainder of that bank will be in the range between that point and approximately 45°C. Hence, the three values of

H_h apply, respectively, to the first and first part of the second section, the last part of the second, and the whole of the third. Small departures from the indicated temperature ranges will not affect the values of H_h to an appreciable extent, so that with reasonable accuracy they may be considered as constants.

Let the condenser now be considered as divided into four sections of which 1 is the first bank, 2 is the first part of the second, 3 the last part of the second, and 4 the whole of the third bank. Then, since the first is in counterflow, the function of H_c and H_h is $\frac{1}{H_h} - \frac{1}{H_c}$, while the sign changes to plus for the second bank and back again to minus for the third. Calling this function H_o we may compute the following values

Section	H_o
1	0.580
2	0.703
3	0.121
4	0.0247

Coefficients of heat transfer. The heat transfer coefficients q_h and q_c for this size pipe and spacing and the corresponding mass flows of gas and air per unit area are computed from equation (3). As already determined, or computed from previously determined dimensions and gas flows, the values to be used are as follows:

	Exhaust Gas above 53°C.	Exhaust Gas below 53°C.	Outside Air
Cp	0.26	2.2	0.24
M, grams per sec. per sq. cm.	1.19	1.19	2.43
μ, poises	2.4×10^{-4}	2.0×10^{-4}	1.85×10^{-4}
D, cm.	2.54	2.54	5.93

Substituting these values in equation (3)

q_h = 0.0012 above 53°C.
= 0.0090 below 53°C.
q_c = 0.0014

Hence the numerical values of $\frac{q_h\, q_c}{q_h + q_c}$ for the sections numbered 1 to 4 above are as follows, Q_o representing the function of q_h and q_c

Section	Q_o
1 and 2	0.000647
3 and 4	0.001211

All the functions of q_h, q_c, H_h, H_c, and p in equations (1) and (2) may now be combined in a single list of constants by multiplying together H_o, Q_o, and p. Calling this constant K,

Section	K
1	0.00299
2	0.00362
3	0.001169
4	0.000239

Equations (1) and (2) now take the form

$$\log_e \frac{T_a - t_b}{T_b - t_a} = KL \text{ for counterflow}$$

and

$$\log_e \frac{T_a - t_a}{T_b - t_b} = KL \text{ for parallel flow}$$

For T_a, the exhaust gas temperature at entrance to the condenser, 650°C. may well be taken. This approximates the melting point of aluminum and should be a safe figure as regards damage to ill-cooled parts of the header casting. For t_a, the initial air temperature, 15°C. is probably a fairly high figure.

With these assumptions, only two temperatures and the length remain as unknown quantities. The only way to arrive at the desired final exhaust temperature, T_b, is to assume a length and apply the equations for each section, finding the final T_b for the last section. The length of the condenser may then be increased or decreased accordingly as the final temperature of the exhaust gas is too high or too low.

For this purpose an effective over-all length of 20 feet has been assumed, making a total length of 60 feet in three banks, of which the second is parallel flow. Keeping to the same designation of the four sections involved in the calculations, we have for Section 1

$$\begin{aligned} L &= 20 \text{ ft.} = 610 \text{ centimeters} \\ T_a &= 650°\text{C.} \\ t_a &= 15°\text{C.} \\ K &= 0.00299 \end{aligned}$$

$$\log_e \frac{T_a - t_b}{T_b - t_a} = KL = 0.00299 \times 610 = 1.824 = \log_e 6.20$$

$$\frac{650 - t_b}{T_b - 15} = 6.20$$

$$t_b = 650 - 6.20T_b + 93.0 = 734.0 - 6.20T_b$$

Simultaneously

$$(T_a - T_b)H_h = (t_b - t_a)H_c$$

or

$$\begin{aligned} (650 - T_b)1.56 &= (t_b - 15)16.08 \\ t_b &= 15 + 63.1 - 0.0971T_b \\ &= 78.1 - 0.0971T_b \end{aligned}$$

Therefore

$$\begin{aligned} 743.0 - 6.20T_b &= 78.1 - 0.0971T_b \\ 6.103T_b &= 664.9 \\ T_b &= 108.9°\text{C.} \end{aligned}$$

The conditions at Section 2 are thus determined:

$$\begin{aligned} T_a &= 108.9°\text{C.} \\ t_a &= 15°\text{C.} \\ T_b &= 53°\text{C.} \\ K &= 0.00362 \end{aligned}$$

Since parallel flow now prevails, the main equation becomes

$$\log_e \frac{T_a - t_a}{T_b - t_b} = KL = 0.00362L$$

$$\log_e \frac{108.9 - 15}{53 - t_b} = 0.00362L$$

$$\log_e \frac{93.9}{53 - t_b} = 0.00362L$$

and also $(T_a - T_b)H_h = (t_b - t_a)H_c$

$$(108.9 - 53)1.56 = (t_b - 15)16.08$$

$$t_b = 15 + \frac{1.56}{16.08}(108.9 - 53)$$

$$= 15 + 5.42 = 20.42$$

substituting above for t_b

$$\log_e \frac{93.9}{53 - 20.42} = 0.00362L = \log_e \frac{93.9}{32.58}$$

$$\log_e 2.88 = 1.058$$

$$L = \frac{1.058}{.00362} = 292 \text{ cm.}$$

The length for Section 3 will be the difference between 610 and 292 centimeters. The conditions at this section are then

$$T_a = 53°\text{C.}$$
$$t_a = 20.42$$
$$L = 610 - 292 = 318 \text{ cm.}$$
$$K = 0.001169$$

The flow is still parallel as in Section 2 so that

$$\log_e \frac{T_a - t_a}{T_b - t_b} = KL$$

$$\log_e \frac{53 - 20.42}{T_b - t_b} = 0.001169 \times 318 = .3717 = \log_e 1.450$$

$$\frac{32.58}{T_b - t_b} = 1.450$$

$$t_b = T_b - 22.46$$

Simultaneously

$$(T_a - T_b)H_h = (t_b - t_a)H_c$$
$$(53 - T_b)17 = (t_b - 20.42)16.08$$
$$t_b = 20.42 + 56 - 1.056T_b$$

Dropping t_b

$$T_b - 22.46 = 76.42 - 1.056T_b$$
$$2.056T_b = 98.88$$
$$T_b = 48.1°\text{C.}$$

The conditions for Section 4 are now determined as

$$\begin{aligned} T_a &= 48.1\,\text{C}^\circ \\ t_a &= 15^\circ\text{C.} \\ L &= 610\text{ cm.} \\ K &= 0.000239 \end{aligned}$$

The flow is now changed again to counterflow so that

$$\log_e \frac{T_a - t_b}{T_b - t_a} = KL$$

$$\log_e \frac{48.1 - t_b}{T_b - 15} = 0.000239 \times 610 = 0.1458$$

$$= \log_e 1.157$$

$$t_b = 48.1 - 1.157T_b + 17.35$$

$$= 65.45 - 1.157T_b$$

Simultaneously

$$(T_a - T_b)H_h = (t_b - t_a)H_c$$

$$(48.1 - T_b)11.5 = (t_b - 15)16.08$$

$$t_b = 15 + 34.4 - 0.716T_b$$

$$= 49.4 - 0.716T_b$$

Hence

$$65.45 - 1.157T_b = 49.4 - 0.716T_b$$

$$0.441T_b = 16.05$$

$$T_b = 36.4^\circ\text{C.}$$

This temperature is somewhat higher than that required for 100 per cent condensation under the assumed conditions, but the difference is no greater than might be caused by the probable error in the assumed values used in computing the heat transfer. The fact that the cooling surface of the cast aluminum headers has not been taken into consideration makes probable a somewhat better performance than that indicated by the computations.

Construction

It is not assumed that a three-bank condenser of the type designed and built by the Bureau of Standards is the ultimate

solution of the problem of exhaust heat transfer. Many other designs may be conceived that will have as good or better performance. As a rule, the construction of the airship to be equipped will have a decided influence upon the condenser design. It is probable that at some future date a metallic or fabric section in the outer envelope of rigid airships could be used to advantage, at least for the later stages of cooling.

In any construction using drawn tubes and cast aluminum headers, the joint of the tube with the header is apt to present a problem. One satisfactory solution has been found in pressing or driving the tubes into holes reamed in nipples cast on the headers. These holes have parallel sides for a short distance and then taper slightly to a shoulder whose depth equals the thickness of the tube.

Expansion. Care must be given the matter of differential expansion between banks of a multi-bank condenser as well as between tubes in a given bank. The former can be taken care of by the provision of freely moving slip-joints between banks at the ends opposite the return bends connecting adjacent banks. In order to avoid the ill effects attending differential expansion between various tubes within a bank, it is necessary to so design the headers and return bends that approximately equal resistance to gas flow is encountered in all tubes.

Separators. The importance of the separator for the collection of entrained moisture in the cooled exhaust gas cannot be over-emphasized. No matter how nearly air temperature is reached in cooling the gas, much of the water condensed remains in the form of spray and is carried along by the gas until the separator is reached. In the absence of such a device this entrained moisture represents a loss of about 50 per cent of the water condensed.

A separator designed for a condenser unit to serve a 300 horsepower engine is shown in Figures 103 and 104. Its action depends upon the drops of water being thrown against the

baffles shown in Figure 103 and guided by the small vertical gutters to holes punched through the bottom to a false bottom or reservoir from which the collected water is drained to the ballast tanks.

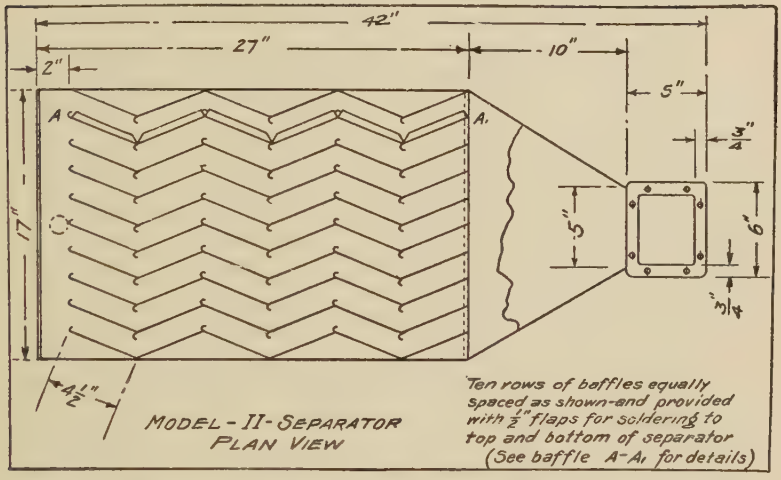

Figure 103. Separator Designed for a Condenser Unit to Serve a 300 Hp. Engine

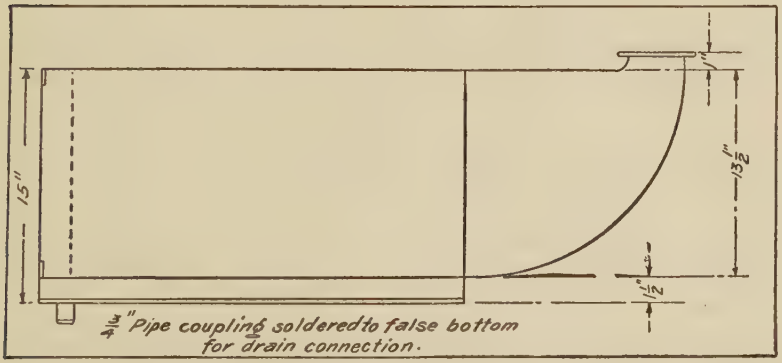

Figure 104. Separator Designed for a Condenser Unit to Serve a 300 Hp. Engine

The exhaust gas enters at the upper right (Figure 104) passes between the baffles and escapes at the left in the same

direction as that of the air stream. The lip on the outlet end is provided to prevent the separated water washing along the bottom past the drain holes and so escaping with the gas.

In designing a separator, it is well to keep toward the lower limit of gas velocity. The limiting values are about 900 and 500 feet per minute, but 600 has proved satisfactory. Below 500 feet per minute the velocity is too low to throw out the water, while above 900 the drops deposited on the baffles are again torn off and return to the gas stream.

Drainage. It is of the utmost importance that all drains for condensed water be large and smooth and free from shoulders or obstructions of any kind. The water condensed is always contaminated with more or less carbon which has a tendency to clog drain pipes wherever a lodging place presents itself.

Mechanical. In order to avoid the possibility of blowing the headers off the tubes due to the explosion of unburned mixture in the condenser, as well as to prevent any tendency of the headers to creep off the tubes due to vibration, it is well to provide steel tie wires parallel to the condenser tubes and connecting the headers to each other. These wires must be designed with regard not only to strength, but to elasticity, since the linear expansion of the tubes is enough to stress them severely as long columns if this is not taken care of by stretching of the wires.

In coupling the condenser to the engine of an airship, the greatest care must be employed to avoid the transmission of engine vibrations, as well as eliminating excessive stresses on the connecting parts. It is also essential that a condenser be rigidly anchored against fore-and-aft motion, only at the end at which the exhaust gas enters. The other end must be free to allow for the over-all expansion and contraction of the unit.[1]

[1] A thorough discussion of Ballast Water Recovery including test results and detailed description of apparatus may be had in Air Service Information Circular (Aerostation), Volume I, No. 44, May 1, 1924.

The heat transfer involved in water recovery is discussed in *Industrial and Engineering Chemistry*, Volume 16, No. 9, page 885, September, 1924, under the title "Heat Transfer in the Condensation of Water from Engine Exhaust Gas."

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